HEATING AND VENTILATING

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Reference Numbers

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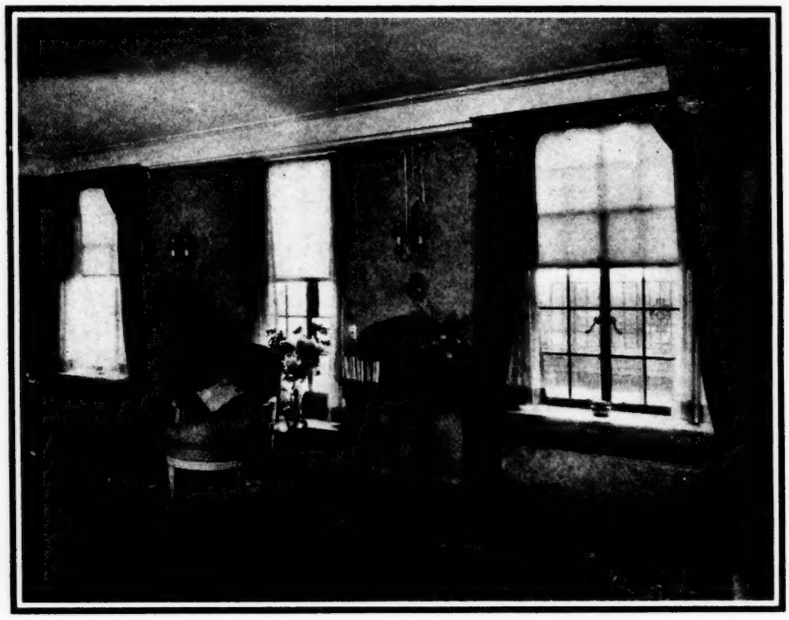

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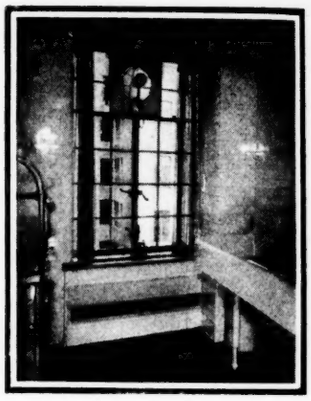

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Editorial

Air Conditioning

THE term "air conditioning" has received much publicity during recent months. Not only has it come into increasingly common use among those professionally identified with it, but the general public has become more widely informed through its numerous applications to theatres and department stores, and through advertising and editorial material which has recently appeared in magazines having huge circulations.

This increasing attention and widening knowledge on the part of the general public should be a source of deep satisfaction to all of us in the profession and in the industry. The use of processed air is no new conception to us. Most of us must certainly have realized at least some of the desirable possibilities which the widespread use of such air may be expected to bring about, not only from the standpoint of our business but from the social and economic sides as well. The rather rapid rise of the term to a place in the popular vocabulary should be wholly desirable. The increased knowledge should lead to increased demand and everyone should be happy.

The very fact that the term is coming into glib use so swiftly gives rise to possible dangers. Recall that this is not the first term or idea to be suddenly seized upon by the general public. Indeed, every few years see the rise of one of them. Very often the fall is just as rapid.

Perhaps the most tragic example and one well known to engineers is the term "efficiency." Who of us does not recall the open-handed welcome which Taylor's announcements on shop management received, not only from engineers but from the press and public? The word was soon on everybody's tongue. No sooner was this true than the "efficiency engineer" came along. True, he might know little of Taylor and less of factory management but he did know the more high-sounding words of the trade. He could talk glibly about wage differentials and time studies even though he was just a bit hazy about what they meant. True, he did not usually last long individually. What he did do was to last long enough collectively to give the word "efficiency" as applied to industrial management such a black eye that it disappeared from the popular vocabulary about as rapidly as it came in.

There is a lesson in this for all of us. Let us all be especially careful during this rather critical period when the public is just forming its impressions that they may be good impressions. One poorly engineered job, or one outstanding failure may be the cause of wholly wrong impressions. Let there be no mushroom growth of self-styled "air conditioning engineers" merely because the term has suddenly found popularity and can be sold. Because a man knows the terms "effective temperature" and "humidity" it does not follow that he is a qualified expert. Yet there are indications that this is exactly what is happening.

Processed air does indeed appear to be on the verge of rapidly expanding applications. It is the duty of everyone interested not to rush headlong into its use but rather to take notice of its promise and to prepare himself to be an "air conditioner" in earnest and not merely in name. Fortunately, the more basic information is easily acquired. Unfortunately, the details of application, so essential to successful operation, are relatively difficult to get. If the word continues to be popular and if it can be sold, lack of ability is not likely to deter many unscrupulous people from capitalizing on its popularity. Those of us who have its long-period development at heart should not lose our balance over what may simply turn out to be a passing flurry of popular interest. What we should do is to help it to develop into a long-sustained interest. Now, if ever, we need balance, sound thinking and, above all, sound engineering so far as the use of conditioned air is concerned.

A New Method of Estimating Steam Consumption

THE degree day unit was originated for use in estimating fuel consumption in buildings. The essential assumption which makes the use of such a unit possible is that the fuel consumption, no matter in what terms it is stated, varies directly with the difference in the temperature maintained in the heated space and the temperature in the outside air. Obviously, such an assumption is not entirely correct, for the fuel consumption is certainly a function of many variables in addition to this difference in temperature. Experience seemed to show, however, that while these other variables do affect the consumption, the temperature difference is the greatest, or rather the governing, variable. It is evident, therefore, that the degree day unit is necessarily an approximation.

It has been evident to those who give the subject close thought that refinement in the estimate can be introduced if the estimated figure warrants so doing. Since in most cases an approximate estimate is all that is required, most people using the degree day as an estimating unit have not found it necessary to introduce any refinement which would complicate the calculation somewhat.

However, one prominent manufacturer of a proprietary heating system has found it desirable for his purposes to apply more exact methods into his estimating procedure. As a result this organization has developed a very interesting, distinctive and copyrighted method of calculation of the heat requirements using the degree day as a basis. The principal points in this method are here outlined.

It should be pointed out that attainment of steam consumption within the heating requirements as estimated by this method is dependent on installation and use of means permitting control, from a central point, of heat delivery to all radiators.

To show the difference between two methods of calculating let us assume that a certain building in New York is to be heated by steam and that its calculated heat loss, as arrived at by the usual methods, is 240,000 B.t.u. per hr. This would normally call for the installation of 1000 sq. ft. of equivalent cast iron radiation.

USUAL METHOD OF CALCULATION

Assuming that each square foot of radiation condenses ¼ lb. of steam per hr., when the inside temperature is 70° and the outside temperature is 0°. Assuming a day during which the outside temperature is 0° for the 24-hour period and the inside temperature is 70° during the 24-hour period, the steam which would be condensed would be $24 \times \frac{1}{4}$ or 6 lb. per sq. ft. for the 24-hour period.

If the base used for calculating degree days is taken at 70°, we would have 70 degree days during that day. In that case the steam consumption per square foot per degree day would be 6/70 lb. However, if the day had had 35 degree days (on the 70° base), instead of 70 degree days, our steam consumption would have been one-half as much as on the 70 degree day.

Assume that the steam consumption can be as readily estimated over the seasonal period as over a daily period, if the length of the heating season in days and the average outside temperature are known. The calculation (for New York) would be as follows:

Length of heating season = 270 days.

Average outside temperature = 40°.

$$270 \times (70 - 40) \times \frac{6}{70} \times 1000$$

= 694,000 lb. for the building when we maintain 70° inside during the whole heating season.

While this method has been widely used for estimating steam consumption, there are at least two difficulties at once evident. One is the fact that very rarely are buildings heated to 70° during the entire heating season, night and day. The second is that measurements taken in the field have shown that the consumption is not truly proportional to the difference in temperature. Therefore, estimates arrived at by the use of this method are very often in error.

Very early in the development of the degree day as a distinct unit it was found by some investigators that for residence work, using gas as a fuel, the proportionality between the outside temperature and the fuel consumption become much more nearly a straight line function when the base is set at 65°.

As a consequence, it has become common practice to set up the degree day unit for practical use on the basis of 65°. The degree days appearing in the tables in the *Heating and Ventilating Degree Day Handbook* are presented in this way.

Based on a 65° base, our calculation would be

$$5348^* \times \frac{6}{70} \times 1000 = 457{,}000 \text{ lb.}$$

per season.

The discrepancy is, of course, accounted for by the difference between the 8100 and 5348. This is due to the fact that in figuring our mean seasonal temperature we have included all days during which the temperature was above 65°, and in addition subtracted the mean from a higher base, i.e., 70° rather than 65°, whereas in the degree day method we have used only the days during which the temperature was 65° or below.

Thus it will be seen that the degree day unit on the usual 65° basis takes into account, to some extent, the fact that the heat loss is not exactly proportional to the outside temperature

**From Degree Day Handbook, for New York.*

and compensates somewhat for the fact that the maintained temperature is not 70° throughout the heating season.

THE NEW METHOD OF CALCULATION

Now let us assume the same building and follow the method of calculation used by the manufacturer who has developed it.

Step 1. Determine the date of beginning and end of the heating season, and the number of elapsed days between these dates, the number of degree days (65° base, from the *Heating and Ventilating Degree Day Handbook*) for that locality for an average season, the average wind velocity, and the average temperature maintained inside during occupancy. For our example we will then have:

Item (a) Heating season started October 1 and ended July 1 a total of

Item (b) 270 elapsed days.

Item (c) 5348 degree days (65° basis) from *Handbook*.

Item (d) 12 miles per hour actual average wind velocity.

Item (e) 70° maintained inside temperature *during occupancy*.
Note—this does not necessarily mean 70° maintained inside during the whole day.

Step 2. Correct the tabular degree days (65° basis) for the maintained temperature and the prevailing wind velocity, as follows:

Item (f) [Item (e) — 65°] × Item (b)
= [70° — 65°] × 270
= 1350 degree days corrected for 70° temperature.

Item (g) = [Item (c) + Item (f)] × $\frac{\text{Item (d)}}{100}$ =
[5348 + 1350] × $\frac{12}{100}$
= 804 degree days corrected for wind

Item (h) = Item (c) + Item (f) + Item (g)
= 5348 + 1350 + 804
= 7502 degree days as adjusted for both inside temperature and wind.

Step 3. Determine the total computed heat load to be supplied, including the heat loss from piping, in terms of the equivalent cast iron direct radiation, recording the temperatures and wind velocity used in the calculation.

Item (i) Radiation 1000 sq. ft.

Item (j) Other radiation, such as pipe coils, in terms of equivalent cast iron (240 B.t.u. per sq. ft.) Radiation—None.

Item (k) Total calculated radiation, 1000 sq. ft.

Item (l) Total installed radiation, 1000 sq. ft.

Item (m) Excess of installed over calculated radiation—None.

Item (n) Radiation from exposed piping, equivalent direct radiation, sq. ft.—None.

Item (o) Radiation from covered piping, equivalent direct radiation, 100 sq. ft.

Item (p) Total equivalent radiation load. Sum of above item, 1100 sq. ft.

Item (q) Minimum outside temperature used for heat loss calculation, 0°.

Item (r) Inside temperature used for heat loss calculation, 70°.

Item (s) Wind velocity used for heat loss calculation, 15 m.p.h.

Step 4. Calculate the heat requirements in B.t.u. per sq. ft. of equivalent direct radiation per adjusted degree day.

Item (t) Assumed Temperature—Wind (T–W) factor = 3*

Item (u) Estimated unit heat requirement base = [Item (r) — Item (q)] + [Item (s) × Item (t)] = [70° — 0°] + [15 × 3] = 115

Item (v) = $\frac{\text{24 hours} \times 240}{\text{Item (u)}}$
= $\frac{24 \times 240}{115}$
= 50 B.t.u. per sq. ft. of equivalent direct radiation per adjusted degree day (with 24 hour operation).

Step 5. Estimate the average weekly operating rate of the building. The operating rate is the ratio of the rate of operation actually required in an average or typical week due to the occupancy conditions to be met, to the rate of operation that would be required in a similar average week if the occupancy were 24 hrs. per day with continuously maintained inside temperature.

Item (u) Average weekly operating rate assumed as 50%.

Step 6. Calculate the seasonal heat requirements in B.t.u. under 24 hour operation.

Item (x) = Item (v) × Item (h)
= 50 B.t.u. × 7502
= 375,100 B.t.u. per sq. ft. per season.

Step 7. Calculate the seasonal heat requirements per sq. ft. of required radiation on the basis of the average weekly operating rate.

Item (y) = Item (x) × Item (w)
= 375,100 × 50%
= 187,550 B.t.u. per sq. ft.

Step 8. Calculate the seasonal heat requirement of the total required radiation.

Item (z) = Item (k) × Item (y)
= 1000 × 187,550
= 187,550,000 B.t.u.

Step 9. Calculate the seasonal heat requirements of the piping. (Assume that the piping is hot 10 hours per day).

Item (zz) = Item (o) × Item (b) × 240 × 10
= 100×270×240×10
= 64,800,000 B.t.u.

Step 10. Calculate the total heat required per season.

Item (A) = Item (z) + Item (zz)
= 187,550,000 + 64,800,000
= 252,350,000 B.t.u.

Step 11. Calculate the pounds of steam required (970 B.t.u., the latent heat per pound of steam at atmospheric pressure)

Item (B) = $\frac{\text{Item (A)}}{970}$
= $\frac{252,350,000}{970}$
= 260,500 lb. per season

It will be noted that the steam consumption as calculated is quite different from the results obtained by the other methods of calculation, in spite of our not including piping in the first two calculations and also in spite of the fact that the number of degree days for the heating season has been increased by the corrections. The decrease is largely due

*An experience factor: derivation explained later in this article.

to the more accurate account taken of the method of operation of the heating plant. It will be noted that short hour heating is assumed in the example, since the estimated operating rate is 50%.

EXPLANATION OF TEMPERATURE WIND FACTOR

The method is comparatively simple except for three points which need explanation. The first is in calculating Item (g). The degree day load corrected to 70° inside temperature is multiplied by the prevailing wind velocity divided by 100. This amounts to an allowance of 1% increase in seasonal heating load for each 1 mile per hour prevailing wind velocity. This factor is based on experience to date with the operation of numerous buildings well distributed geographically and is subject to revision as added experience is obtained.

The second point needing explanation is the so-called temperature wind factor, Item (t). This is also an experience factor.

A mathematical analysis for approximately evaluating this constant can also be used. It consists of assuming that for each radiator installed a certain portion is installed to take care of the loss from the building due to conducted heat losses in assumed still air, and the remainder is necessary to supply the loss due to all air movement, including infiltration. Analysis of the heat loss through the walls and windows of office buildings of ordinary construction shows that the infiltration losses are in the neighborhood of 39% of the total heat loss.

Since the temperature difference in our example under designed conditions is 70°, and since the temperature accounts for the difference between the total heat load and the 39% for air movement, then the 70 units represent 61% of the total load. Therefore, 70 divided by 0.61 would mean that 115 units would represent the total load. Consequently, we must add 45 to our 70 to account for the infiltration, so that we multiply our wind velocity by some figure which will give us 45. Since our wind velocity in the heat loss calculations (not in the heating load) is 15, the factor necessarily must be 3.

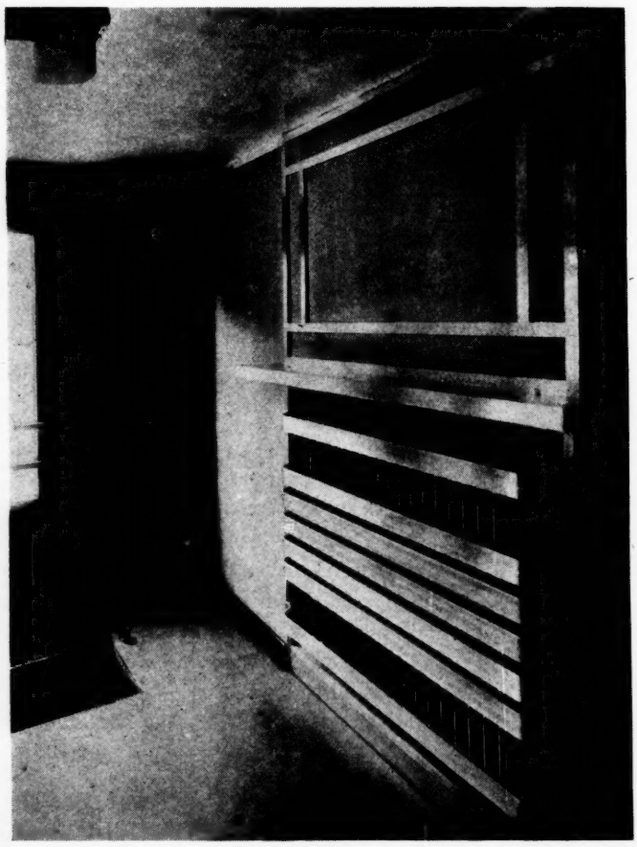
The third point concerns the method of operation or "operating rate." The method of estimating described provides for determination of an operating rate which takes into consideration (1) Maintaining the desired inside temperature only during the normal daily or weekly period of occupancy, (2) Allowing suitable daily heating-up periods during which heat delivery will be in excess of that required for normal maintenance of temperature, (3) Provision of means for controlling, from a central point, heat delivery to all radiators and, (4) Utilization of such control means to assure that delivery of heat will be substantially in proportion to need therefor.

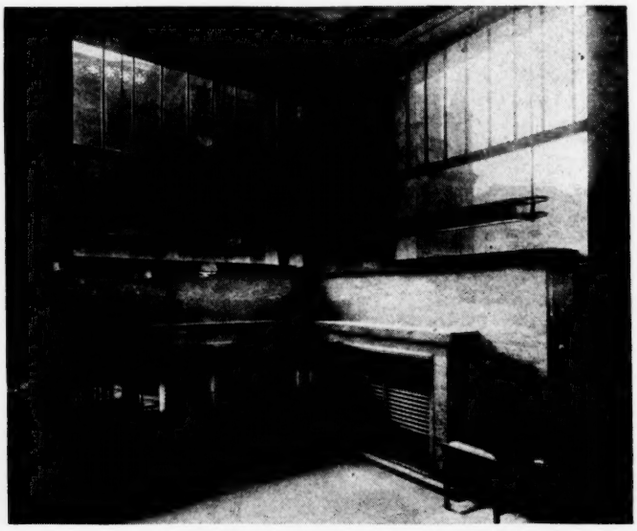
LIMITATIONS OF METHOD

One limitation of the method must be mentioned. This is that use of the method presupposes that all of the radiators in the building are assumed as being supplied with steam. In the case of hotels, where the number of radiators which are shut off due to non-occupancy of the rooms may be extremely high, the method consequently is not applicable.

To summarize, while the preceding explanation has been somewhat lengthy, it is in the end rather simple. It takes into account the following factors with which we were not concerned in our first two calculations. These are:

1. Actual inside temperature maintained or to be maintained.
2. Wind.
3. The operating rate or method of operation.
4. The heat loss from the piping.





Interior Views of Branch Post Office on the Unter-den-Linden, Berlin. Left, the vestibule, with ornamental radiator grilles of bronze. Below, bronze radiator grilles in the booking office.

Photos by Arthur Pakuscher, courtesy Copper & Brass Research Assocn.

A Study of the Combustibility of Anthracite

by **R. V. Frost***

ONE accustomed to the burning of anthracite or bituminous coals is not unfamiliar with the aggravating annoyance caused by a certain variability in the burning qualities of the fuel. What may be the cause of this variability does not seem to be well understood by either the producer or the dealer, and certainly not by the consumer.

Coals in the anthracite field have long been classified in a rough way as hard or free burning, but never has a real attempt been made to thoroughly investigate the subject. As a result, rule-of-thumb methods of coal selection have come into practice. Now a coal dealer buys his coal largely by appearance. If the coal is shiny and black he considers it a better coal than one that is dull and somewhat gray. Some dealers will take only coal that is cubical in form and others buy only fresh mined coal, notwithstanding it is a point of economy to have coal delivered in the early summer, in which case some of the coal has been in the open for at least a year by the time it is used.

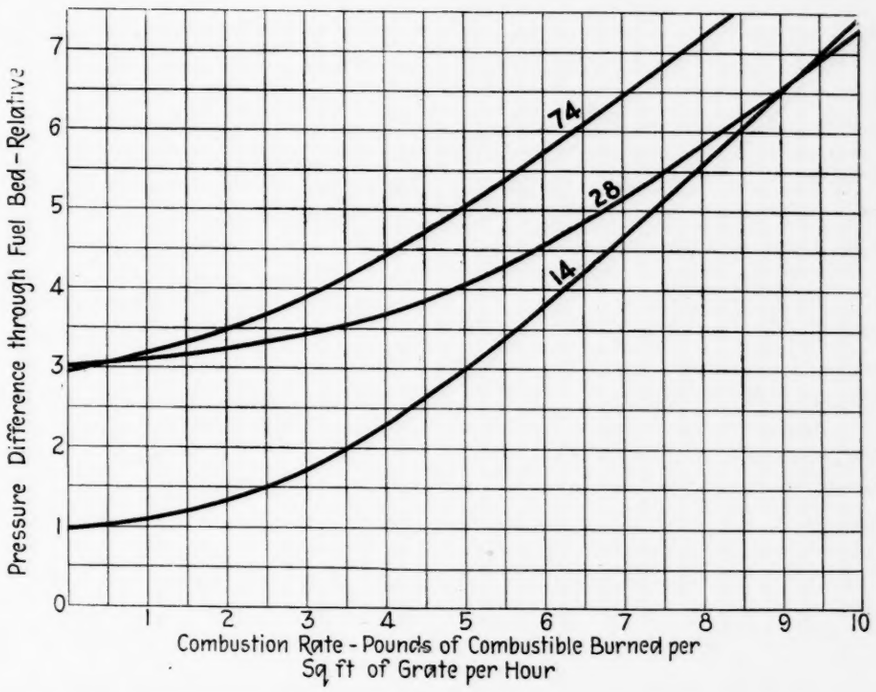
All of this is a hit or miss fashion of buying and not at all comparable with the methods of classification used in the purchase of iron ore, oil, steel, copper or any other of the basic commodities. The producers have no method of telling whether the coal they are shipping is free or hard burning. The coal loaded in one car may be very free burning while the coal loaded in the next car may be as hard burning as can be found.

Variability in the combustibility of both anthracite and bituminous is not only annoying to the domestic consumer, it is economically a serious matter to the industrial consumer and particularly so in stoker equipped plants. In the small stoker field it undoubtedly is seriously retarding the development of automatic coal burners.

The variability in combustibility in fuels makes itself evident in the difference in the amount of draft required to burn different coals at a given rate of combustion and also in the different rates at which different coals will ignite in the fire bed.

It is not at all uncommon to obtain coals in the anthracite field that require, in order to burn at the same rate of combustion, two or three times the amount of draft required of another; or, one coal at the same draft will burn five or six times as fast as another. From the ignition standpoint some coals will, at the same combustion rate, ignite seven or eight times as fast as others, or some coals must be burned at three or four times the rate of combustion of others in order to get the coal to ignite as quickly.

The difference between hard and free burning coals would not be so serious were a consumer able to obtain continuously just the one grade of coal, but when one month he has delivered to him a shipment of free burning coal and the next month hard burning, he is up against a serious problem. If the fuel is burned in a stoker all sorts of complications ensue. Every time the coal is changed the air must be readjusted to the rate of coal feed and frequently the stoker has not been designed for a hard burning fuel. Conversely, in the case of a free burning coal, the stoker may not have been designed for that grade of coal, the result being a loss of heat due to excess of air. These troubles cause some coals to be condemned for use in some stokers and have even forced certain coals off the market entirely, so that they are not used for any purpose, either hand or stoker fired.



Draft Differential Variation with Combustion Rate Chemical Analysis:

Sample No.	74	14	28*
Percent Moisture	.....	1.0	2.8
Percent Volatile	.....	6.8	2.5
Percent Fixed Carbon	.....	82.9	66.8
Percent Ash	10.0	9.3	27.9
B.t.u. per Lb.	13,210	13,310	10,250
Specific Gravity	.....	1.38	1.68

**Raw Coal Sample.*

Fig. 1. Rate of Combustion Plotted Against Draft

**Director, Frost Research Laboratories.*

Frequently there is a change in the combustibility of the coal in a single shipment and in such a case the change results in loss of capacity as well as in decreased efficiency and when used in a power plant the operator has no means of immediately putting his finger upon the trouble.

Could the consumer buy his coal on the basis of a combustibility ratio, somewhat after the manner in which oil is purchased, no complaints because of troubles with hard and free burning coals would be heard, for equipment could be adjusted to burn one grade and as long as the consumer obtained that grade satisfaction would result.

It is easy to see that variation in combustibility can cause no end of trouble in boiler selection. Undoubtedly many of the complaints about boiler ratings and poor chimneys can be attributed to the fuel used. Consequently, the heating boiler industry, in the protection of its own interests is vitally concerned in the subject of fuel combustibility and its correct understanding.

The average anthracite mine works from six to ten veins and while some of the veins may be free burning coals others can be extremely hard burning. The method of mining in most mines necessitates loading a train of mine cars, usually about ten, from a given level, elevating them to the top of the breaker and passing that coal through the breaker in one unit for the various processes of washing and sizing. Thus the coal from one particular vein is isloated and finds its way into one car with very little possibility of blending with coals from other veins of the mine.

While the output of many anthracite mines is quite uniform in combustibility, many others are not so satisfactory and contain deposits that run to extremes in variation.

Blending of the coals from different veins is recognized as the correct solution of the difficulty, but unless the blending is controlled by some intelligent method it is hardly worth the bother. Some of the mines attempt to blend, but there is in these attempts apparently no well defined method of control.

A study of the combustibility of fuels has lead to the development of a method of test that enables one to determine with accuracy the true combustible nature of the coal. Typical charts showing the results of such tests are shown in Figs. 1 and 2. In Fig. 1 drafts required to maintain given rates of combustion are plotted, and in Fig. 2 there is plotted the rate of horizontal fire travel against rate of combustion. Three different coals have been plotted to illustrate the range of variability in the coals, these being just about the extremes of those found in the anthracite field. Samples No. 14 and No. 74 were obtained from dealers' yards, while No. 28 is a sample procured at the mine before preparation for the market. No. 14 is a free burning type, while No. 74 is hard burning.

The samples originated from different mines, but it is possible to secure from the same mine coals as widely variable as are these samples. In fact, samples taken separately from the top and bottom of the same vein in the same locality are often found to vary to an even greater extent than these. But if all the samples from each of the mines from which these two samples were taken, were carefully blended and then sampled it would be found that the blended coal from one mine is very much like the blended coal from the other, so closely does the combustibility of coals average.

A study of the charts shows that to burn these coals at a 5 lb. rate of combustion, the amount of draft required for No. 74 is to the draft required for No. 14 as 5 is to 3, and at a 3½ lb. rate, which is normal burning rate in domestic heating plants, the ratio is 4 to 2. Taken in another way, No. 14 will burn at the rate of 5 lb. per sq. ft. of grate per hr. on the same draft that will barely start No. 74. Fig. 2 shows that No. 74 required a combustion rate of 18 as against 3 for No. 14 in order to get horizontal ignition started, and that No. 14 will burn horizontally at a ratio of 13.4 as against 1 for No. 74 when burning at equal rates of combustion.

From the foregoing it is seen how clearly the difference in the combustibility of the coals can be indicated.

The method of test is practical and simple in operation, requiring only an understanding of the fundamentals of combustion to get accurate results. Most of the apparatus used is standard laboratory equipment, the determinations being made in two miniature furnaces of special design, carefully calibrated to secure the desired accuracy.

Determinations are based on close measurement of air flow, air pressure, rate of combustion, accurate sizing in the preparation of the samples, control of ignition and measurement of temperatures of the burning fuel.

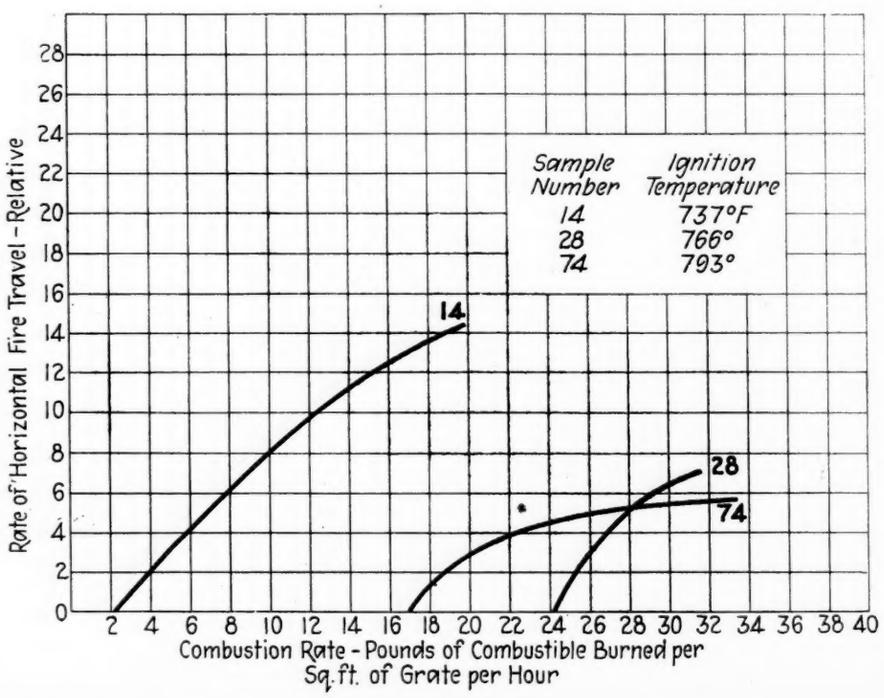


Fig. 2. Rate of Combustion Plotted Against Fire Travel

A Method of Attacking Certain Problems in Industrial Air Conditioning

by **Arnold Weisselberg***

Part II

THE saturation curve is called the boundary curve because it divides the chart into two regions: the region of superheated vapor above and the region of watery vapor, or fog, below. We are, therefore, in a position to follow the changes of state into the region of fog.

Let us assume that we have air of a state denoted by point D in Fig. 3, with a moisture content of 0.0245 lb. and a heat content of 41 B.t.u. per lb. If we subject this air to a cooling effect beyond the dew point, the state point enters the region of fog, with part of its vapor condensing into minute particles of moisture, which remain afloat in the air unless given an opportunity to precipitate. Suppose that the cooling is continued until point D_2 on the 65° isotherm is reached. The amount of heat that has to be removed in order to reach this state is found to be 41 — 22 = 19 B.t.u. per lb. of mixture.

If we remove the condensed vapor by precipitation, the change in state will take place along the line D_2D_3 since the temperature remains constant. Point D_3 has a heat content of only 21.7 B.t.u., hence 0.3 B.t.u. per lb. was carried away with precipitation. Point D_3 is reached after the moisture removed amounts to 0.0136 lb. per lb. air, which is the scaled distance of point D_3 from the vertical DD_2.

State D_3 may also be reached by other methods than that set forth above. One of these is accomplished by adsorption followed by cooling. The change in state due to adsorption takes place along the isotherm line to the left, because the vapor removed has, according to the steam tables, a heat content of 1100 B.t.u. per lb. and it will be found that the direction of the isotherm is parallel to the direction of the 1100 B.t.u. line of the marginal scale. We note that, in order to reach state D_3 by cooling following the adsorption process, the adsorption has to be carried to point D_4, whence a cooling effect must be applied to the extent of 27.7 — 21.7 = 6 B.t.u. per lb. of air.

Still another way to obtain air of state D_3 with an initial air of state D would be by mixing with air of another state. Point E denotes an air of 50° F. and 80% relative humidity. If we mix the two kinds of air the resultant mixture will be somewhere on the line DE which passes through D_3. Evidently state D_3 will be obtained if the quantity of air of state E is to the quantity of air of state D as DD_3 is to ED_3.

The last method is typical application of the most salient feature of this chart, namely the graphical solution of problems involving the mixture of two quantities of air of different states. Obviously, it does not make any difference if the one state is in the region of fog while the other is in the region of superheated vapor, the mixing will still take place along the line connecting the two states. For instance, state D_3 can also be arrived at if the foggy air of condition D_2 is mixed with air of state denoted by F in a ratio $\frac{FD_3}{D_2D_3}$. In point D_3 the fog will have disappeared. We are, therefore, in a position to determine by means of this chart the minimum amount of unsaturated air of a certain state required to lift the fog of a known state. It will be observed that if heat is added to the unsaturated air, the ratio will change, showing a reduced amount of unsaturated air to dissipate the fog.

Many processes involve the mixing of two amounts of air of different states. That such a mixing takes place is not always apparent, and only a closer analysis will prove it. The process of evaporation under atmospheric conditions at temperatures below the boiling point is such a case. The evaporating water does not, as would appear, escape into the surrounding air, and become absorbed by it, but what actually takes place is a mixing of the air with a thin layer of saturated air above the surface of the water and of the same temperature as the water.

With this in mind and applying Dalton's relation $E = N(e' - e)$, in which E is the rate of evaporation per unit time and area, N, a constant, for a constant set of conditions, e' the vapor pressure at the temperature of the water and e, the partial pressure of the vapor in the air (dew point pressure), the process of evaporation under every possible condition can be graphically analyzed by the help of this chart. Evidently the investigation must be based on uniform conditions.

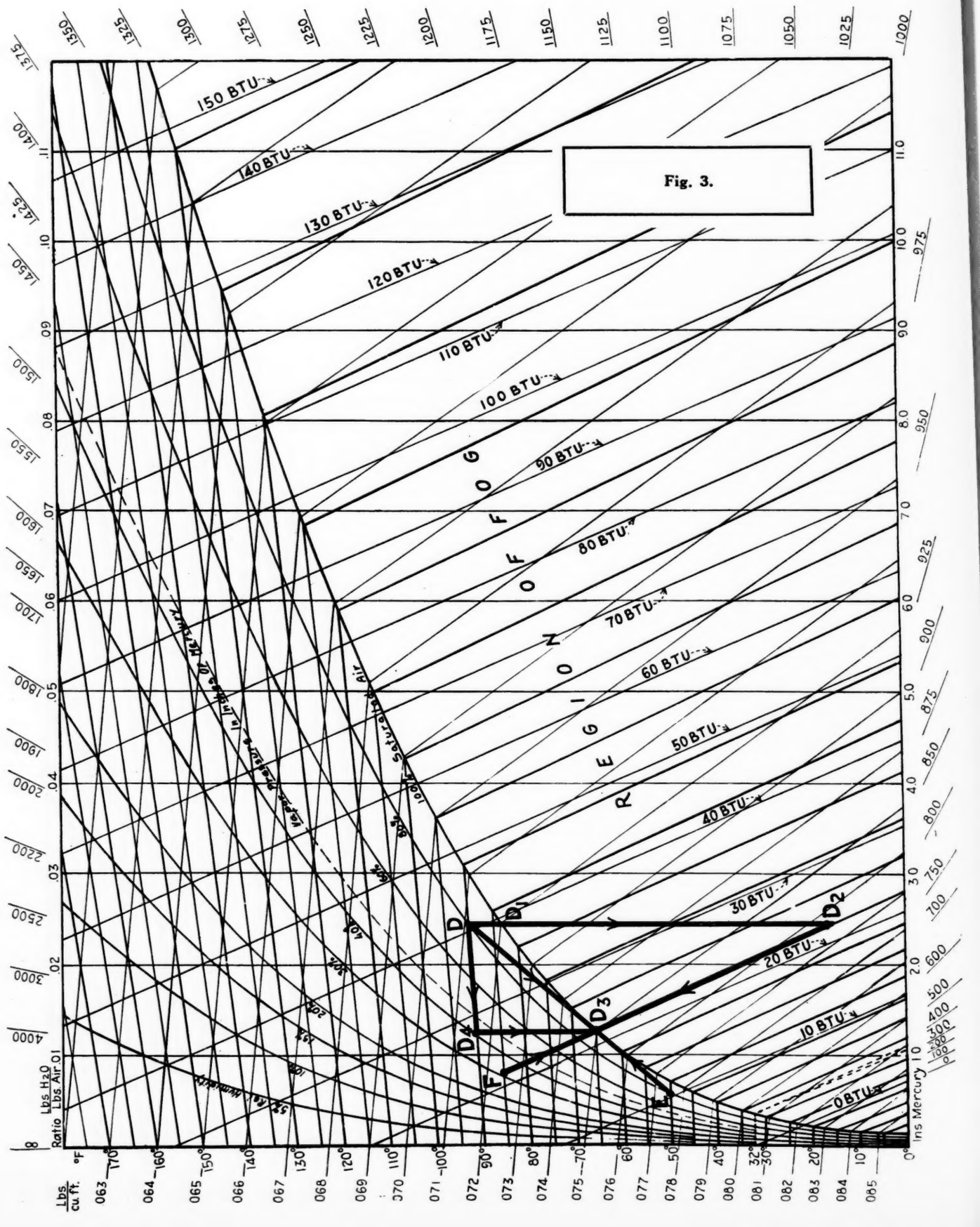
Let us assume the case of a dye house in which the water in the vats is kept at 120° F. The state of the layer of saturated air forming above the water surface is designated by point L in Fig. 4. Further assume that the outside air has a temperature of 80° F. and 70% relative humidity as indicated by point M. This air, upon entering the dye house becomes heated and rises to the ceiling, reaching a maximum temperature a short distance below the ceiling of say 100° F. and 37% relative humidity as designated by point N. The mixing action will take first place with the lower strata of air in the direction of line LM. This line cuts through the region of fog, indicating fog formation.

The foggy air, being lighter than the surrounding air, will rise, reaching warmer strata. The direction in which the mixing occurs will grad-

*Consulting Engineer.

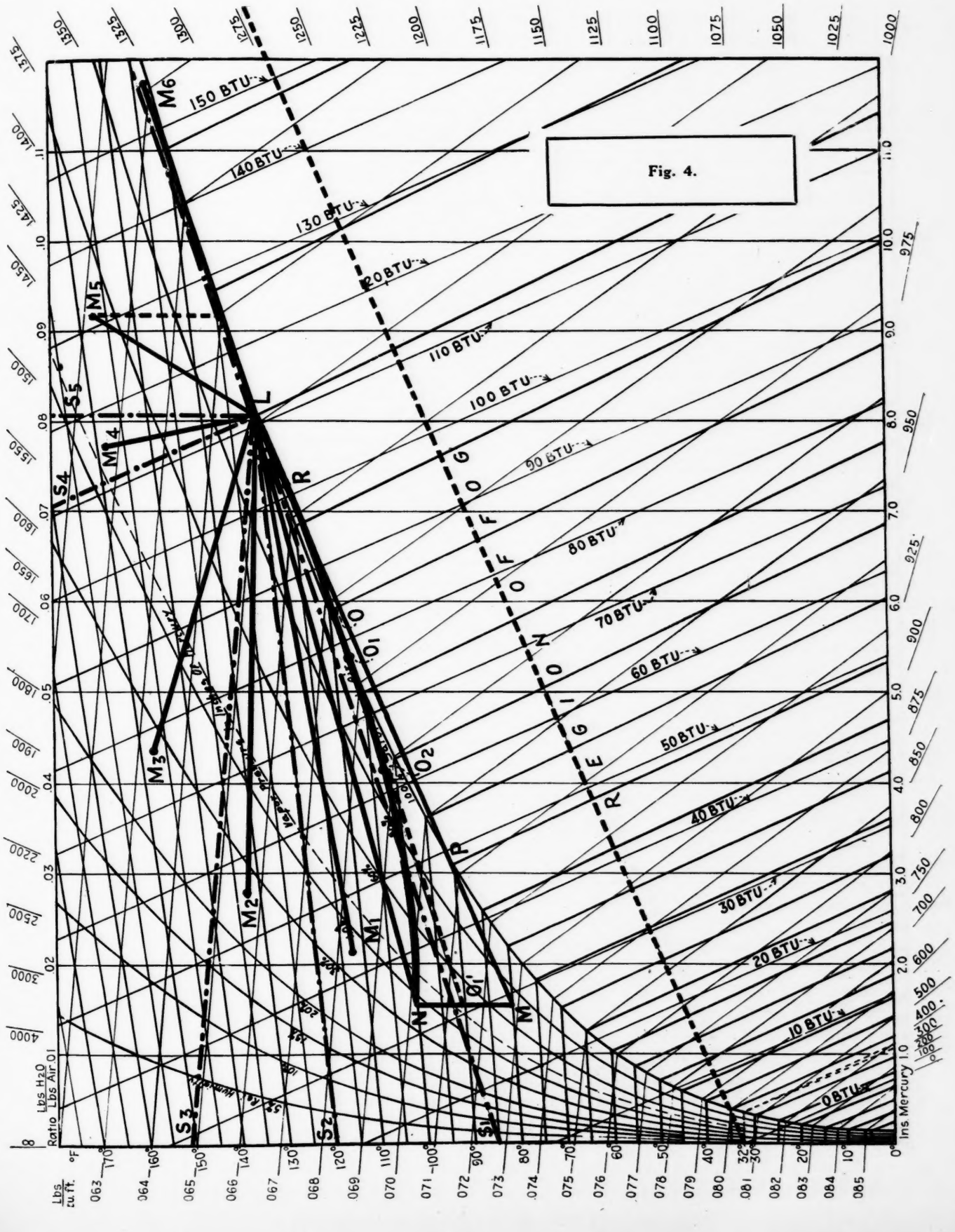
ually change, finally pointing toward N. Let us assume that the conditions are such that the mixing follows the curved path shown between L and N. The state of the mixture at various level heights from the floor will be indicated by points along this path. If the proportion of air admitted and heat liberated in the dye house to the amount of water evaporated is small, the final mixture state, even for the highest level, will still be within the region of fog (point O), i.e., the entire dye house will be wrapped in fog.

Suppose, however, that enough air is admitted to reach point O_2. The intersection with the boundary curve



is at O. The tangent at this point intersects the line NM in point O'_1, indicating the height at which the fog will disappear, if MN represents the height of the room. Obviously, if no fog should occur at the lower levels, as is usually desired, Point O_1 must coincide with point P on the intersection of the line LM with the boundary curve. For this to happen it follows that the amount of air supplied at the lower level must be to the amount of saturated air forming above the water surface at least as LP is to MP.

The amount of saturated vapor forming is a function of the amount of water evaporated and this, in turn,



takes place, as has been shown, in accordance with Dalton's relation $E = N (e' - e)$. In this equation N is really the product of three constants, $N_1 \times N_2 \times N_3$. N_1 is a factor depending on the physical properties of the substances involved in the act of evaporation, such as the respective molecular weights, specific heats and thermal conductivities, N_2 is the factor of resistance to evaporation depending on the condition of the water surface, i.e., whether it is clean or covered by a film of another substance, and N_3 is a diffusion factor, depending on the rate at which the diffusion takes place. N_3 is indicative of the state of the air immediately above the surface of the water, since the formula could be written also $E = N_1N_2 (e' - e_2)$ in which e_2 will vary between the limits e and e', according to the rate of diffusion. With increasing rates of diffusion, i.e., with increasing velocity of the air brought into contact with the water surface, e_2 grows smaller reaching a minimum with e, which is identical with setting a value of 1 for N_3. Going in the other direction, e_2 becomes equal to e', which is equivalent to $N_3 = 0$, i.e., the evaporation comes to an end.

Evidently the existence of a saturated layer of air above the water surface is purely hypothetical and serves only as a basis for reasoning. On the other hand, it does not make any difference, according to the chart, whether the layer of air above the water surface is of the state L or R, as long as it is somewhere on the line connecting L and M, depending on the rate of diffusion: nor will it change the answer as to the ratio required between the amount of air of state M and the pounds of water evaporated, if the fog should be dissipated. What will change is the total amount of air required, since the evaporation will increase if the air above the water surface is in condition R instead of condition L. This is an important conclusion in that it shows the consequence of permitting the water surface to come into contact with air at high velocities. Not only is the heat supply to the vats considerably augmented with every additional amount of water evaporated, but the amount of air that has to be supplied to eliminate the fog has to be increased as well.

All the above is of little import to the practical engineer who is in search of an answer to the question: How much air do I actually require to eliminate the fog? What good is it to know the ratio if we do not know how much water is being evaporated? Actual conditions are never such that they will permit the calculation of the amount of water evaporated from the amount of water surface exposed to the air, even if the rate of evaporation for existing conditions were known. However, the amount of heat, usually in the form of steam, that is supplied to the vats is usually either known or measurable. Evidently, if the temperature in the vats is to remain constant, the heat supplied with the steam must equal the heat escaping with the vapor, if we ignore the small losses caused by radiation and conduction. Then, if we know the amount of heat escaping with a pound of vapor, it is easy to figure the pounds of water evaporated from the amount of steam supplied. Naturally, maximum figures will be used as a basis for calculation. In an existing plant the steam demand can be readily determined by meter. For a proposed plant we will have to assume a figure, based on past experience, of so many pounds of steam per hour per vat.

In calculating the amount of water evaporated from the heat supplied with the steam, we must keep in mind, however, that the heat given up with 1 lb. of water evaporated is more than the heat contained in a pound of vapor, since the air is being heated also. To find the amount of heat escaping with 1 lb. of vapor we again make use of the chart, remembering what was said in connection with the explanation of the marginal scale. Referring again to Fig. 4, draw through the origin of the marginal scale a parallel to the line LM and read on the marginal scale that with every pound of water evaporated, 1271 B.t.u. are carried away, if the water in the vats is kept at 120° F. and the state of the air is as denoted by point M.

We can now proceed to solve the following problem: It is desired to know how much air of state M has to be supplied to a dye room in which 10,000 lb. of saturated steam at 15 lb. gauge are being fed to the vats hourly. Since the heat of such saturated steam is 1163 B.t.u., the answer figures out as follows:

$$\frac{10{,}000 \times 1163}{1271} \times \frac{1}{0.0805} \times \frac{LP}{MP}$$

$$= 360{,}000 \text{ lb. per hr. or } \frac{1}{0.073} \times \frac{1}{60}$$

$$= 89{,}000 \text{ cu. ft. per min.}$$

Let us turn to the chart again. We note that if the state of the air denoted by point M were above the tangent s_1 on the boundary curve in point L, fog would not form under any circumstances, since the connecting line LM_1 does not cut through the region of fog. If the original state of the air should be above the isotherm s_2 passing through L, as denoted by point M_2, the temperature of the air drops in the course of mixing, while the temperature of air of state below s_2 would increase. Air of a state above s_3 (point M_3) will be lighter than the product of mixing and there will be no tendency to set up a natural circulation. A state to the right of line s_4 (point M_4) means that the air will give up heat which evidently must be absorbed by the water. This does not necessarily mean that the water becomes warmer, because the heat removed by evaporation may exceed that given up by the air. However, if the state is at the right of s_5 the water is bound to become warmer since evaporation has ceased and moisture is being precipitated from the air.

One will observe that not only is the moisture content of the air being reduced while the initial state M_5 changes towards L, but that the dew point pressure "e" of the air of state M_5 is greater than e' corresponding to L. This means that $(e' - e)$ in Dalton's relation becomes negative or that the process of evaporation is reversed, which is identical with condensation. Finally, we may have air of a state as denoted by M_6 which, being below the tangent, leads to the conclusion that condensation spoken of in connection with state M_5 will take place under formation of fog.

As greater use is made of this chart no doubt many more applications will be found. With this new contribution Professor Mollier has again placed the engineering profession in his debt.

Air Conditioning

by **William Hull Stangle**

VII—Air Distribution and Duct Friction

by **Realto E. Cherne***

THE subject of air distribution both in straight ventilation work and air conditioning, is one of prime importance because of the possibility of introducing objectionable drafts or dead air pockets. The procedure in laying out a distribution and return system is not one that may be readily placed in a formula or other concise method, due to the nature of the variables encountered. Such factors as ceiling height, length of flow, location of exhaust or return grilles, temperature differential between air entering and room temperature desired, and dispersion of heat loads, all enter into the calculations, and each carries a greater weight than the average designer is wont to admit. In cases where winter-time heating alone is desired, the type of distributing system will be substantially different from a case where cooling is required.

In other words it is impossible to state a general rule without giving the exceptions to the rule; such being influenced by building exposures, lights, occupants, and location of direct radiation, if any. After the location of outlets, return or exhaust grilles and main headers has been ascertained, it is a comparatively easy matter to size the duct system.

The primary consideration in supplying air to a room is to convey the air as directly as possible at a velocity which will best economize space, material and power. Friction loss through elbows and sharp turns should be minimized to save power in operation and to prevent improper distribution which may be caused by resulting eddy currents and turbulence. It is evident, in general, that the greater the ratio of duct perimeter to cross-sectional area, the greater will be the frictional loss.

The most convenient method of presenting data on designing distributing and return or exhaust systems is to take an actual problem

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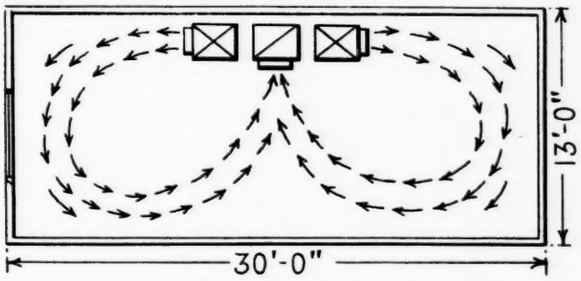


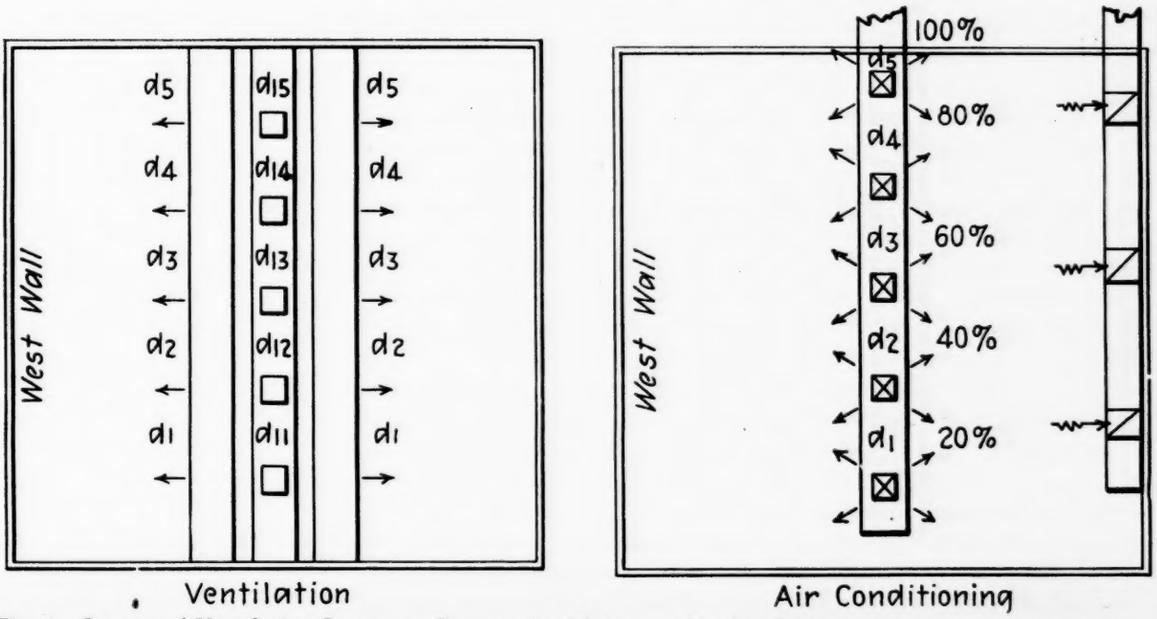
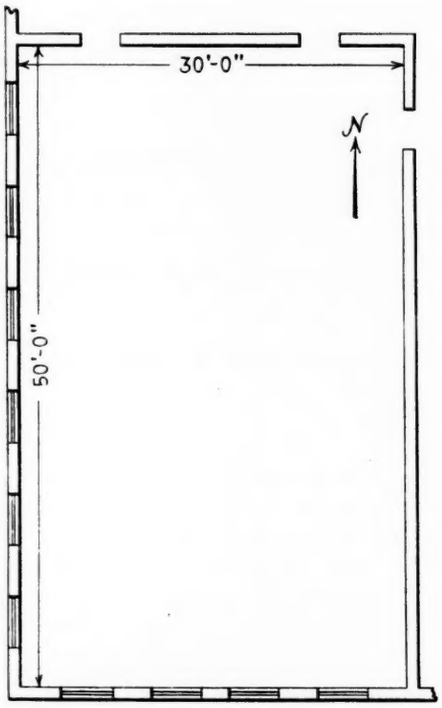


Fig. 1. Layout of Ventilation System as Compared with Layout for Air Conditioning System for Same Room

to solve for both straight ventilation and air conditioning.

Let us assume a S.W. corner room as shown in Fig. 2. For the sake of simplicity we will assume that the rooms above and below and on North and East sides are conditioned or ventilated, as the case may be, and that there will be no heat effect from them on the room being considered. The total floor area of the room is 50 ft. x 30 ft. or 1500 sq. ft. The cubical contents are therefore 1500 sq. ft. x 13 ft. or 19,500 cu. ft. For the purposes of a good ventilating system, we can assume a straight ten minute air change and thereby obtain an air quantity to be supplied to the room of 1950 c.f.m. If a positive pressure is maintained there will be some loss of the supplied air outward which means that the maximum air we can expect to exhaust is about 85% of 1950 or 1660 c.f.m. We will, therefore, design a system to supply 1950 c.f.m. and exhaust 1660 c.f.m.

Before proceeding with the layout, it might be well to digress in order to sum up the heat gains in the room in an effort to find what the temperature rise of the supply air will be under maximum conditions. The maximum heat gain on a room of this exposure will occur at about 3:30 to 4:00 o'clock in New York when the sun is shining directly on the large West wall glass area. In all probability, some of the lights will be on in the dark corner due to the fact that they were needed early in the day and were not turned off. Adding together the heat effects which occur simultaneously due to sun, people, radiation, and lights, we get approximately 340 B.t.u. per min. This gives a gain in sensible heat of about one B.t.u. for each 5.73 c.f.m. of air supplied. Since one B.t.u. will raise 1 c.f.m. of standard air about 56°, the greatest rise to be encountered for the ventilating system under consideration will be 56/5.73 or 9.76°. The maximum conditions, in all probability, will not occur more than a few times each summer, but since we are dealing with a complicated variable it is difficult to estimate the number of times the rise in temperature will approach 10°. Suffice, then, to state that average operating conditions will not be confronted with such a heat



Ceiling Height, 13 ft. Clear; Windows, 4 ft. x 7 ft.; Lights, ½ Watt per sq. ft. over ½ the floor area; Occupation, 1 person per 50 sq. ft. office space; Coefficient for Wall Construction, 0.277; Coefficient for Ceiling and Floor, 0.251.

Fig. 2. Room which is to be Ventilated and Air Conditioned

gain. The above figures are applicable to ventilation but are not strictly true for air conditioning where a definite temperature is being maintained regardless of outside weather variations.

It is customary to size the ducts in such a manner that total pressure drop will not be excessive or go beyond the efficient operating range of the fan. Several methods may be used for calculating sizes, depending upon job conditions.

1. An initial velocity may be assumed which will give the duct size at the outset after which the velocities at various points may be approximated and areas subsequently figured. Unless the designer has had quite some experience, this method will result in uneconomical use of iron or, perhaps worse, objectionable distribution.

2. Assume a maximum resistance which will be allowable and work out the sizes accordingly. In general, this method will give the most economical use of iron, and since the branches will be properly proportioned will insure the required distribution without excessive use of dampers.

3. Perhaps the most convenient method of proportioning ducts in main headers and branches, one in which the calculations are not laborious, is to proportion for equal friction per foot of length. This method greatly simplified the friction calculation since it is necessary only to know the length of the longest run, number and size of elbow, and diameter and velocity in the largest pipe. The friction loss for the system will be the same as though the entire air were being carried the whole distance through the largest pipe. For the purpose of calculating by this method, the following formula may be used:

$$\frac{d_2}{d_1} = \left(\frac{Q_2}{Q_1}\right)^{\frac{3}{8}} = \left(\frac{V_2}{V_1}\right)^2$$

$$\text{or, } \frac{v_2}{v_1} = \sqrt[5]{\frac{Q_2}{Q_1}}$$

wherein d_1 is the diameter of pipe to convey Q_1 cu. ft. of air per min. with a velocity V_1 ft. per min. and d_2 is the corresponding diameter to convey Q_2 cu. ft. per min. at a velocity of V_2 ft. per min.* The word diameter used above is the equivalent diameter of round pipe to carry the same capacity as a rectangular duct and have the same friction per foot of length.

4. On a job where there is one or more long runs of duct with outlets equally spaced it may be advantageous to lower the velocity at each outlet by an amount sufficient to cause the regain in static pressure to just equal the friction loss of that section. Any method which decreases the velocity of air along the run has several important advantages.

(a) The velocity of the air is utilized in producing static pressure.

(b) A nearly uniform static pressure may be secured in all parts of the line which will greatly aid in getting uniform distribution throughout.

(c) The friction in the smaller pipes is reduced as a function of the square of the velocity.

Refer to Fig. 3 which greatly simplifies calculation for sizing a duct where the outlets are equally

**From page 90 "Fan Engineering," edited by R. D. Madison under the direction of Willis H. Carrier. Published by The Buffalo Forge Co., Buffalo, N. Y.*

spaced in a straight run and each discharges an equal amount of air. For example, take the smallest outlet to be 10 in. diameter with 10 outlets equally spaced ten feet apart. $\frac{L}{d} = \frac{10 \times 12}{10} = 12$. Locate 12 at the base of the chart; read vertically upward for the value of $\frac{D}{d}$ for each section, and obtain:

$d_2 = 1.32 \times 10 = 13.2$ in.
$d_3 = 1.54 \times 10 = 15.4$ in.
$d_4 = 1.70 \times 10 = 17.0$ in.
$d_5 = 1.82 \times 10 = 18.2$ in.
$d_6 = 1.92 \times 10 = 19.2$ in.
$d_7 = 2.05 \times 10 = 20.5$ in.
$d_8 = 2.07 \times 10 = 20.7$ in.
$d_9 = 2.12 \times 10 = 21.2$ in.
$d_{10} = 2.17 \times 10 = 21.7$ in.

Should space or other conditions be unfavorable to the use of round pipes, the diameters shown above must be converted to give suitable rectangular ducts.* For the problem solved above, let us assume the maximum allowable depth to be 12 in. For d_2 then, use 12 in. $\times$ 12 in.; d_3, 17 in. $\times$ 12 in.; d_4, 20 in. $\times$ 12 in.; and so on. For installations where the ducts are exposed and there is no restriction as to depth, a 2:3 ratio should be used to give a comparatively economical use of iron and still be pleasing to the eye.

The formula for equal friction per foot of length is far too unwieldy for ordinary calculations and these friction curves were plotted† to give results well within the accuracies of other factors on the job. For a simple problem, to illustrate these charts, assume a 30 in. main duct is to be used. From this, there is a branch leading off which is to carry 50% of the total air. On the left hand index locate 50% capacity. Follow this line horizontally to the right until it intersects the 30 in. diameter of main pipe line. Vertically above this intersection read 22.8 in. which is the size of the branch pipe.

A chart that will prove very useful in designing ducts as well as in determining the friction of a system

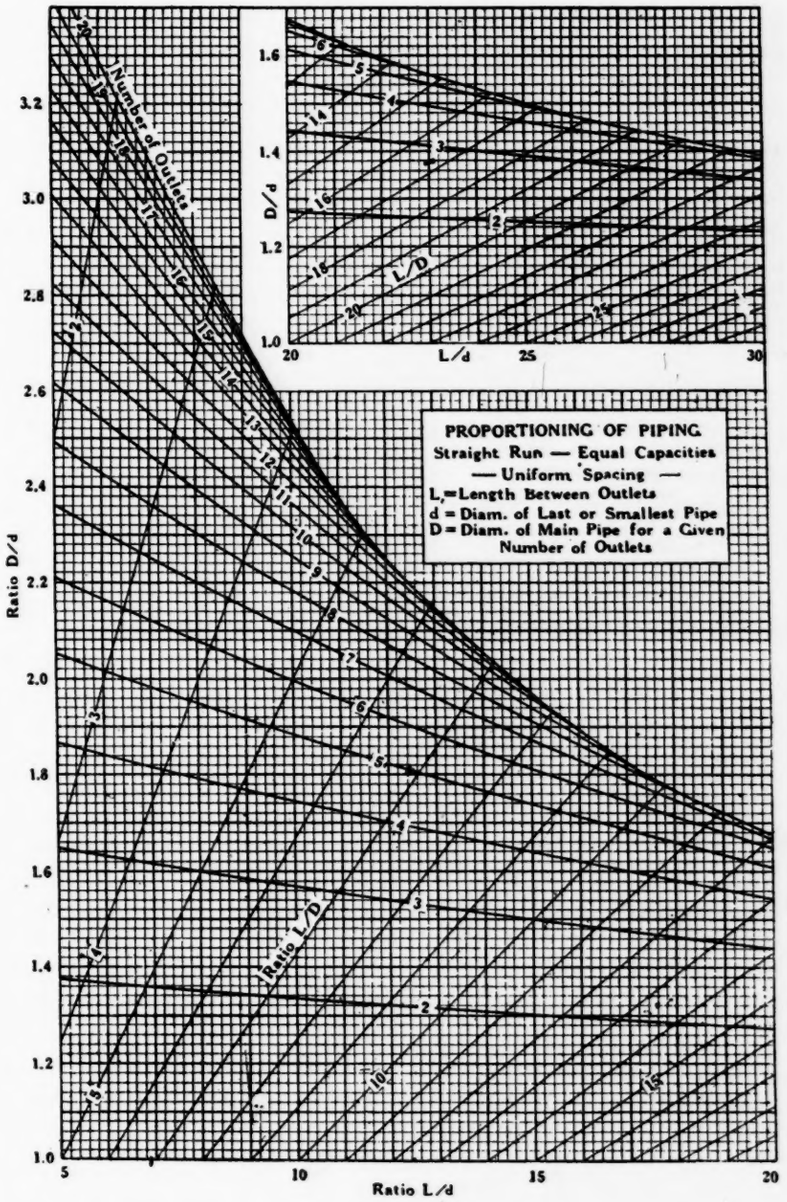


Fig. 3

Reproduced by permission from "Fan Engineering," published by The Buffalo Forge Company.

is given on page 85 of the December issue of HEATING AND VENTILATING. It will be found convenient for determining the friction for a given air capacity in various size ducts at corresponding velocities. The coefficient of friction has been corrected for the various diameters and capacities, based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the capacity. The chart is based on a loss of one velocity head (at a velocity of 2,000 ft. per min.) in a length equal to 50 diameters of 24 in. galvanized swedged pipe.

As an example of the use of the chart, assume that it is desired to pass 10,000 cu. ft. of air per min. through 75 in. of 24-in. diameter pipe. Find 10,000 cu. ft. of air per min. on the right scale and move horizontally left to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3,200 ft. per min. Directly below the intersection it is found that the friction per 100 ft. is 59 in., then for 75 ft. it will be 0.75 $\times$ 0.95 = 0.44 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

The head loss by friction may be expressed as a modification of the Fanning formula: For round pipe and standard air (70° and 29.92 in. mercury):

*Charts showing circular equivalents of rectangular ducts for equal friction can be found in literature of the fan manufacturers, such as "Fan Engineering," in Harding and Willard, etc.

†*See pages 88 and 89, "Fan Engineering," Second Edition.*

$$h = z\frac{L}{D}h_v = \frac{L}{ED}\left(\frac{V}{4005}\right)^2$$

For rectangular duct and standard air this becomes

$$h = zL\left(\frac{a+b}{2ab}\right)h_v = \frac{L}{E}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2$$

where

- h = loss of head in inches of water
- V = velocity of air in feet per minute
- L = length of pipe in feet
- D = diameter of pipe in feet
- h_v = velocity head in inches of water
- a = one side of rectangular duct in feet
- b = other side of rectangular duct in feet
- z = coefficient of friction, varying from 0.01 to 0.03
- E = length of pipe in diameter for one head loss.

The value of E may be taken as 60 for smooth pipe; 50 for ordinary galvanized iron heating and ventilating ducts; 45 for smooth, and 40 for rough conduits of tile, brick or concrete.

Friction coefficients vary with duct size and velocity, as well as with changing air conditions. However, it is needless labor to carefully calculate the theoretical friction losses in a system because of mechanical imperfections incident to fabrication and erection. A rough edge or eddy currents caused by improper construction will throw the actual friction loss far out of line with the theoretical. As an example of the possible variation in these calculations we cite a drying plant for which the total head was carefully calculated and found to be about 1.5 in. water for the entire system. In actual operation, however, the result was about 2 in. of water. Much of this error was due to a series of baffles placed in front of the heater in an effort to insure equal distribution over the entire surface.

Now to return to the solution of the problem of designing a distributing and exhaust system for the room shown in Fig. 2. For ordinary ventilation a good type of distribution would be as is illustrated in the section, Fig. 1. As an illustration it will be sufficiently accurate to supply half of the air from each supply duct, assuming that the light load will tend to counteract, at least in part, the sunlight load on the West wall. This will simplify calculations. If we place the outlets 10 ft. apart we get 10 for a total, which gives 195 c.f.m. per outlet. Better results are obtained with numerous small outlets than with a few large ones. Since the outlets are discharging equal amounts of air and are equally spaced in a straight line, Method 4 given above may be used. Assume an initial velocity of 1000 ft. per min., then

$$\frac{1950}{2 \times 1000} \times 144 = 140 \text{ sq. in.}$$

initial duct area

$$D = \sqrt{\frac{140}{.7854}} = 13.4 \text{ in. diameter}$$

of round pipe to carry 1/2 the air.

Refer to Fig. 3. $\frac{L}{D} = \frac{10 \times 12}{13.4}$

$= 8.95$ (approx.)

Locate 8.95 at the base of the chart and follow diagonally upward to the line of 5 outlets. Vertically below the intersection we read the value of $\frac{L}{d}$, 15.5, from which

$$d = \frac{10 \times 12}{15.5} = 7.74 \text{ in.}$$

$d_5 = 1.72 \times 7.74 = 13.35$
Use 19 in. × 8 in.

$d_4 = 1.63 \times 7.74 = 12.60$
Use 17 in. × 8 in.

$d_3 = 1.49 \times 7.74 = 11.50$
Use 14 in. × 8 in.

$d_2 = 1.31 \times 7.74 = 10.00$
Use 11 in. × 8 in.

$d_1 = 7.74$
Use 8 in. × 8 in.

The depth of the duct was maintained at 8 in. to facilitate carrying an even ceiling height and keeping as much clearance as possible.

Sizing of the return duct is accomplished in a similar manner.

$$\frac{1660}{1000} \times 144 = 240 \text{ sq. in.}$$

$$D = \sqrt{\frac{240}{.7854}} = 17.5$$

$$\frac{L}{D} = \frac{10 \times 12}{17.5} = 6.86$$

Locate 6.86 at the bottom of Fig. 3 and proceed diagonally to the 5 outlet line. Vertically below we obtain

$$\frac{L}{D} = 12.5$$

$$\frac{10 \times 12}{12.5} = 9.6$$

Then

$d_{15} = 1.81 \times 9.6 = 17.45$ in.
Use 34 in. × 8 in.

$d_{14} = 1.69 \times 9.6 = 16.2$ in.
Use 30 in. × 8 in.

$d_{13} = 1.54 \times 9.6 = 14.8$ in.
Use 24 in. × 8 in.

$d_{12} = 1.32 \times 9.6 = 12.7$ in.
Use 17 in. × 8 in.

$d_{11} = 9.6$ in.
Use 10 in. × 8 in.

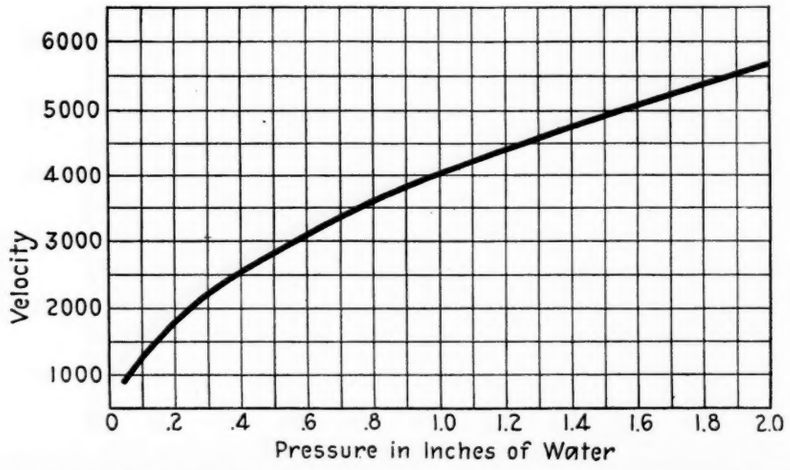


Fig. 4. Velocity in Feet Per Minute of Dry Air at Various Pressures when the Temperature is 70° and the Barometer 29.92 in.

Here, again the rule of conserving clearance was adhered to, although an uneconomical use of iron results.

The friction loss for this system is very easy to calculate hence it is necessary only to determine the losses up to the first outlet. (As was explained elsewhere.) On the curve shown on page 85 of the December issue of this publication locate 975 c.f.m. on right hand index, follow horizontally to the left to the intersection of the 1000 ft. per min. velocity line. Vertically below the intersection read 0.145 in. of water friction loss per 100 ft. The friction, therefore, for 5 ft. of duct is 0.05 $\times$ 0.145 = 0.00725 in. Now, consider the same problems from an air conditioning standpoint. The total maximum heat gain in the room (sensible heat only) will be approximately 390 B.t.u. per min. instead of 340 as given for straight ventilation because a definite temperature will be maintained which requires that the radiation inward must be provided for. (Assume 80° dry bulb and 50% relative humidity to be maintained in New York climate).

The temperature differential between entering cold air and desired room temperature with the ceiling height and blow available and using pan outlets may approach 12° under maximum load conditions. Therefore the total air to be supplied will be approximately 1820 c.f.m. which is slightly below the amount used for the 10 min. air change ventilation system. A good system of distribution will result if a supply duct is run down the ceiling at the center of the room and the return along the east wall with drops to grilles near the floor. The outlets, again, will be placed 10 ft. on centers which results in 5 outlets discharging 365 c.f.m. each. Three returns of 550 c.f.m. each will be ample.

To illustrate Method 3, this system will be sized for equal friction per foot of length.

Assume an initial velocity this time of 1200 ft. per min. and obtain the initial duct size of 16.66 in. diameter. d_1 = 20% = 8 in.; d_2 = 40% = 11.5 in.; d_3 = 60% = 13.5 in.; d_4 = 80% = 15 in.; d_5 = 16.66 in. If a 10 in. maximum depth is maintained d_1 = 6 in. $\times$ 10 in.; for d_2 use 11 in. $\times$ 10 in.; for d_3 use 15 in. $\times$ 10 in.; for d_4 use 19 in. $\times$ 10 in.; for d_5 use 4 in. $\times$ 10 in. The return should be sized in similar manner.

The velocities used in the above problems are by no means correct for good distribution and quiet operation, but merely serve for illustration. Many times the velocity is dictated by space conditions or other factors.

Up to this point nothing has been said of friction in elbows. It has been found that a center line radius of 1½ D for round pipe or 1½ W for rectangular pipe gives good results and that increasing this radius to 2 D or 2 W gives practically no additional gain in power conservation. The friction loss is normally expressed in per cent of velocity head. For an elbow for a round duct having R = 1½ D the loss will be about 17% of a velocity head.*

In ordinary calculations for round pipe with elbows of the proper radius, the friction loss for one elbow may be assumed to be about equal to the loss in 10 diameters of straight duct.

This brief discussion of air distribution and distributing systems serves to familiarize one with the nature of the many variables to be encountered. The best way to become proficient in air conditioning or ventilation layout design is to be confronted with numerous different problems to be solved. These will involve use of the charts given and bring out the fact that the subject of air flow should be understood at the outset.

**See pages 76 and 77, "Fan Engineering."*

Ventilation Important Problem in Proposed Tunnel

THE question of ventilation is the most important problem in the planning of the proposed tunnel to connect Brooklyn and Staten Island, according to Colonel J. R. Slattery, deputy commissioner of the Board of Transportation of New York City. The tremendous stretch of the tubes, which will lie one hundred feet beneath the waters of the channel, will require the forcing of air through them to keep the atmosphere purged of gases left by automobiles. Twelve million cubic feet of air will be pumped through the tubes every minute, making necessary tubes larger in girth than those of the Holland Tunnel.

According to Colonel Slattery, the ventilating system will also be larger than that used in the Holland tunnel because of the greater length of the tubes. There will be vent towers near each bulkhead line and two more inland, one on each side of the Narrows. Fresh air will be forced in beneath the road and sucked out by exhaust fans in the roof of the tubes.

The first step is to find the tentative location of the tunnel and its exits. The next is to determine how much ventilation will be necessary for a given length. Then the size of the ducts must be fixed so that the diameter of the tubes can be calculated. Plans must be made for the approach to the tunnels; ample plazas must be figured on. These decisions having been reached, the rest of the work is easy. While the general project of the Narrows tubes has been settled upon, the details have yet to be worked out.

The tunnel will lie under the Narrows connecting Brooklyn and Staten Island, the Narrows being the gateway through which the major portion of the transatlantic shipping enters and leaves New York harbor. The tubes will be 32 ft. in diameter, exceeding those of the Holland Tunnel by two feet. In length it will be 5,000 ft. between ventilation shafts and 11,000 ft. between portals; corresponding measurements of the Holland tube are 3,500 and 8,000 ft. Present estimates are that the cost of construction will be $78,000,000.

According to present indications, the entrance on the Brooklyn side will be in the vicinity of Eighty-sixth Street, and the Staten Island entrance will be near Hylan Boulevard. The tunnel's chief service will be to by-pass much of the traffic between Jersey on the one side and Long Island and New England on the other, which at present is obliged to push its way through crowded city thoroughfares.

When the Heating Contractor Wakes Up . . .

by **William E. Taylor**

THE author of this article has been for the past nine months Executive Secretary of the Fact Finding Committee of the New York Building Congress, and for three years prior to that Executive Secretary of the Heating and Piping Contractors New York City Association. The organization of the Heating Board of Trade of New York was largely due to his influence.

In this article, and ones which will follow, Mr. Taylor discusses, from the standpoint of one who is close to the contractor, the problems which heating contractors are facing.

WHAT can be said which will be of value in restoring the industry of heating and ventilating to business health?

Along with many other businesses it is sick.

Tobogganing was never a satisfactory sport to me because the climb back took all the joy out of the ride down.

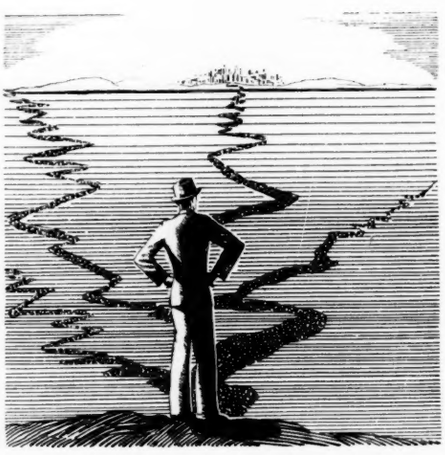
For over twenty years the heating contractor has been operating his business along the lines of least resistance and that always leads down hill.

Well, he has had his ride. He can't go much farther. Is the climb back worth while? Can he do it? What road shall he take?

I think the effort is worth while. A national survey made by the Heating and Piping Contractors National Association, dated June 18, 1930, says that it is worth while to the tune of $500,000,000 a year—this being the value of heating and ventilating work in new construction, not including warm-air furnace work, nor any work in modernizing 16,000,000 homes.

I think he can do it. I have been close to him for over twenty years and have grown to know and respect him. I believe that he has both the brains and the courage required to get back upon the highway of success.

What road shall he take? This is a more serious question. If it were to be answered in the light of his actions thus far I would change one word to make it read: *What road will he take?* The question is then unanswerable because he has always thus far been a true disciple of rugged individualism and, apparently, been willing to pay the price.



What Road Will the Heating Contractor Take?

Individualism means he will do as he pleases, when he pleases, and to whom he pleases, without regard to any resulting effects outside of the immediate transaction.

If I did not believe that he is now ready to harness his individualism, limit its freedom and intelligently direct it, the problem would not be worth bothering about.

Therefore, a review of some of the matters which have contributed to the existing demoralization will be of value in selecting the road back and harnessing individualism with the wisdom needed to avoid a wreck on the way.

Most of the matters to which I shall refer during the series which will follow are known to the trade. That is, a few will be recognized as intimate personal acquaintances by a few, and what they don't recognize will be recognized by others so that,

in the sum total, a complete picture will be presented—a complete specification of things that are bad for the industry, in which all may join in avoiding in the personal conduct of their business for the benefit of their business and the industry.

Make no mistake, your business can be healthy only as the industry is healthy and, conversely, the industry cannot be healthy unless the private businesses promote health.

We have had rather a blistering experience in the past year, and one thing has been made clearer to the average mind than ever before—that all industry is interdependent to maintain prosperity. A heating business controlling purchases of over $350,000,000 a year in materials and equipment and over $150,000,000 a year in wages to labor, touches many industries and is an important factor in the economic life of the country, an additional reason for cleaning house.

The first thing, I believe, that the heating contractor must realize and understand before he tackles this job, is that he has a big job ahead, a job worthy of the best effort, the best brains in the trade. Don't think that it will be anything but a hard, difficult task requiring time. It cannot be done by any short cut, for along that way lies danger. It cannot be done by letting George do it.

If the principle that action and reaction are equal can be applied to your problem the road to recovery will be long, and whether it applies or not, the road back will be long and difficult enough.

The difficulty and time elements need not discourage you for your rewards will be evident as you make progress and there will be much encouragement and satisfaction along the way. You can win only by realizing just what the job means and stop eternally seeking a panacea which will change loss into profit overnight.

That is the reason why the trade association has failed to function: it never had a chance because most of the members of an association take it for granted that by becoming a member and paying the dues, they purchased a miraculous process of relief from the headache so common in the trade.

I shall deal with this trade association angle at some length later. It has an important part to play in the recovery, but the popular conception of its functions must be completely rebuilt. It is a place for work and there is nothing automatic about it.

Old-timers in the business resent the presence of the newcomers and, in many instances, this resentment is justified when considered solely from the viewpoint of the destructive effects of their acts due to lack of experience or the necessity created by lack of financial resources.

A Good Heating Contractor is a Practical Engineering Executive

However, that is not getting at the root of the trouble, which in reality can be traced to the door of the old-timer who, by using short cuts and his bag of tricks, so cheapened his own importance in experience, organization and resources that it became easy for those of the lowest grade in these requirements to procure consideration almost on the basis of equality.

The good heating contractor is more than a contractor, he is a practical engineering executive—the filter through which the theories of design pass to actual creation. In the early days he was just that and secured a large part of his business only by reason of his reputation in this regard.

Then the trouble started. He began by requesting prices from material suppliers, not by specifying the quantity, size and type of material to be bid upon, but by the use of that all-meaning phrase "in accordance with plans and specifications" or "as specified by architect for such and such a job."

This short cut in the contractor's office made it necessary for the seller of material to take off the list of material required from the plans and specifications and become responsible for accuracy of quantity and price.

Thus a train of events was started which discounted experience and ability.

Almost anyone who could add a column of figures was enabled to make up a bid. A postal card addressed to suppliers of the materials specified requesting prices in accordance with plans and specifications brought the prices on a large part of the work, and the labor cost, with other items, was guessed. Anyone could and did go into the contracting business.

Then, too, the supplier came to realize that it was to his advantage to have his price in as many hands as possible so that if he received no request for prices he sent out a bid unsolicited to contractors with whom he desired to develop business relations. The inevitable happened. More and more competition, lower and lower levels of prices, more and more tricks of the moment, an endless tail chasing process leading where?

If that old-timer who knew his business, knew how many valves he required, the size and type, had asked for a bid upon his own quantities and taken the responsibility for accuracy, there would be today fewer so-called heating contractors and those that did exist would be more like the practical engineering executive.

Some will say this is all bunk because the supplier of material would quote others on the quantities required by a responsible contractor or call attention to discrepancy of quantities.

That is true to some extent, but bear this in mind: the seller of material would be glad to be relieved of the expense and responsibility involved in determining the information which is your job and in exchange would adopt the practice of treating your request for price as strictly confidential.

I know that this is true because suppliers of material are already combating the existing practice and want to be relieved, not only to reduce needless expense in their own organizations, but to improve conditions in the trade.

Destruction of this practice can begin at once. There are thousands

of so-called contractors who could not make up an accurate list of materials for a given job. Of course there are more today who could do so than would have been the case if the short cut had never come into use but, even though belated, that door can be shut by determination on the part of good contractors to stop it and by cooperative actions among trade groups.

This is a proper field of activity for the trade association provided the membership want the door shut.

There are other angles to this matter relating to quantities for organized groups but none will in any way interfere with the effectiveness of the contractors' determination to do this job now.

One of the basic reasons for the chaotic conditions in the trade is the attitude of contractors toward each other.

In all relations of life except business the basis of friendship is a mutuality of interest, but when it comes to business those with common interests look upon each other as natural enemies and act accordingly. Almost anything that can be done to injure a competitor is considered as an exemplification of cleverness and the greater the injury done the more satisfaction it gives the doer. As if to complete this distorted attitude they look upon the purchaser of their services as their natural friend and don't seem to realize that in the period preceding contract relations, the time of negotiation, these interests are opposed.

How many times have those who read this taken an unprofitable contract in the belief that it placed them in a favorable position to secure later business upon better terms only to find that they had completed their usefulness in the first transaction by giving the purchaser something for nothing.

They have harmed not only themselves in the loss they have sustained but have proved to the purchaser that he can continue to play the same game with his future business and use a different contractor on each transaction.

How often has the contractor been told by the purchaser that his competitor's bid is so much lower than his and accepted such information as a clear indication of friendship when the facts have revealed that it was not the truth.

It is this unwarranted attitude of childlike trust in the friendship of the purchaser which underlies much of the price-cutting activity and resulting demoralization.

The silly part of it lies in the fact that the contractor has at his disposal effective ways to stop the practice but he won't believe anything a competitor says and has the utmost

The Scenery in the Purchaser's Office Is Intentionally Set to Deceive the Contractor

confidence in anything a purchaser tells him in spite of repeated experiences which have cost him money.

I have personal knowledge of repeated instances in which the so-called successful contractor had taken the contract at a price he knew was below his actual cost of material and labor because he had been told by his friend the purchaser that the bid of his competitor made it necessary for him to quote that price in order to give him the contract, when the truth was that his competitor's price was 20% above that figure.

I know of instances where the scenery in the purchaser's office has been intentionally set by preparing a typewritten contract containing price and all conditions beforehand, anticipating the call of a contractor, it did not matter much which one, who would be permitted to see the drawn contract, accidentally of course, and the bait has been swallowed.

Much is heard in the way of complaint about the practice of shopping bids by general contractors and others when as a matter of fact, no general contractor has to do any shopping in order to get a price to a low enough point. All he has to do is to wait and the contractors will relieve him of the necessity for any action by voluntarily forming in line to seek the chance of giving him a revised price, always revised downward, and thank him for the opportunity and consider that again he has received positive evidence of friendship and preference for association with him as the contractor.

Curses and bad names have been hurled at the general contractor blamed for this practice. Resolutions have been passed by building congresses and trade associations. Nevertheless, the one who is actually responsible for it goes merrily on passing resolutions—and cutting his price as an integral part of his process of doing business.

Sooner or later he finds himself in a hole and then proceeds to parcel out the hole to the people from whom he buys materials, and by that process conveys the infection of unmoral business throughout the structure of his industry, until finally he wonders why his trade has so many damnable conditions to contend with.

There is some hope, however, because his sources of supply are in more compact units. They can figure their costs more easily, they keep better records and they are awakening to the need for establishing a deadline on price beyond which no man shall go. Further than that, they are preparing to eliminate the contractor if necessary in order to sell their product at a profit.

There used to be a legitimate reason for trade protection when the contractor was a source of help to distribute manufactured products, but the necessities of the situation into which the contractor has brought himself by lack of foresight and bad thinking (or none at all) has changed the contractor from a source of help to one of positive interference in the manufacturer's plans and there is less of trade protection in the manufacturer's thoughts today than ever before in the history of the industry.

So from the contractor's standpoint the need to be up and doing is great and I have not exaggerated in the least in the opening paragraphs when I say it is a long, hard job but worthy of the best that the heating contractor can give to the doing of it, systematically and with patience born of a realization that the reward will be worth the effort.

Estimating and Designing Heating Systems

XIV—Two Pipe Atmospheric Pressure Systems

by **Fred W. Hanburger***

THE steam heating layout for the installation illustrated is an atmospheric system for heating a two-story garage 120 ft. deep and 60 ft. wide. It is a replica of an actual layout now being installed, with slight modifications to suit the purpose of this article. This is shown in Fig. 33. The lower floor has a heat loss of 390,000 B.t.u. and is to be heated by four suspended unit heaters. There is also a locker and washroom on this floor supplied with 80 sq. ft. of direct indirect radiation.

The upper floor, consisting of offices and warerooms, has an estimated heat loss of 192,000 B.t.u., and is heated with 10 radiators of 80 sq. ft. of cast iron radiation each. The excess heat loss in the lower floor is due to an added allowance for air change occasioned by the use of large doorways at each end of the garage which are almost continually opened for several hours both morning and evening. There is no excavation underneath the garage and a boiler room is provided beneath the sidewalk.

To keep the drawings as clear as possible, most of the necessary valves, traps, and accessories are not all shown, but are referred to in the text. The mains are carried below the ceiling of the garage proper and the returns along the side walls near the floor. The first step is to determine the size of the unit heaters to be used. Their excellence for heating large rooms has resulted in their increasing use and increased variety of types. The ratio of the fan capacity and the heating surface gives a great variation in the difference between the temperature of entering and leaving air. Some types with an entering temperature of 60° F. will deliver the air at 150° F., and in others, the delivered air under the same condition may be as low as 100°.

Both conditions have their advantages and the choice is left to the designing engineer as to the temperature at which it is desired to have

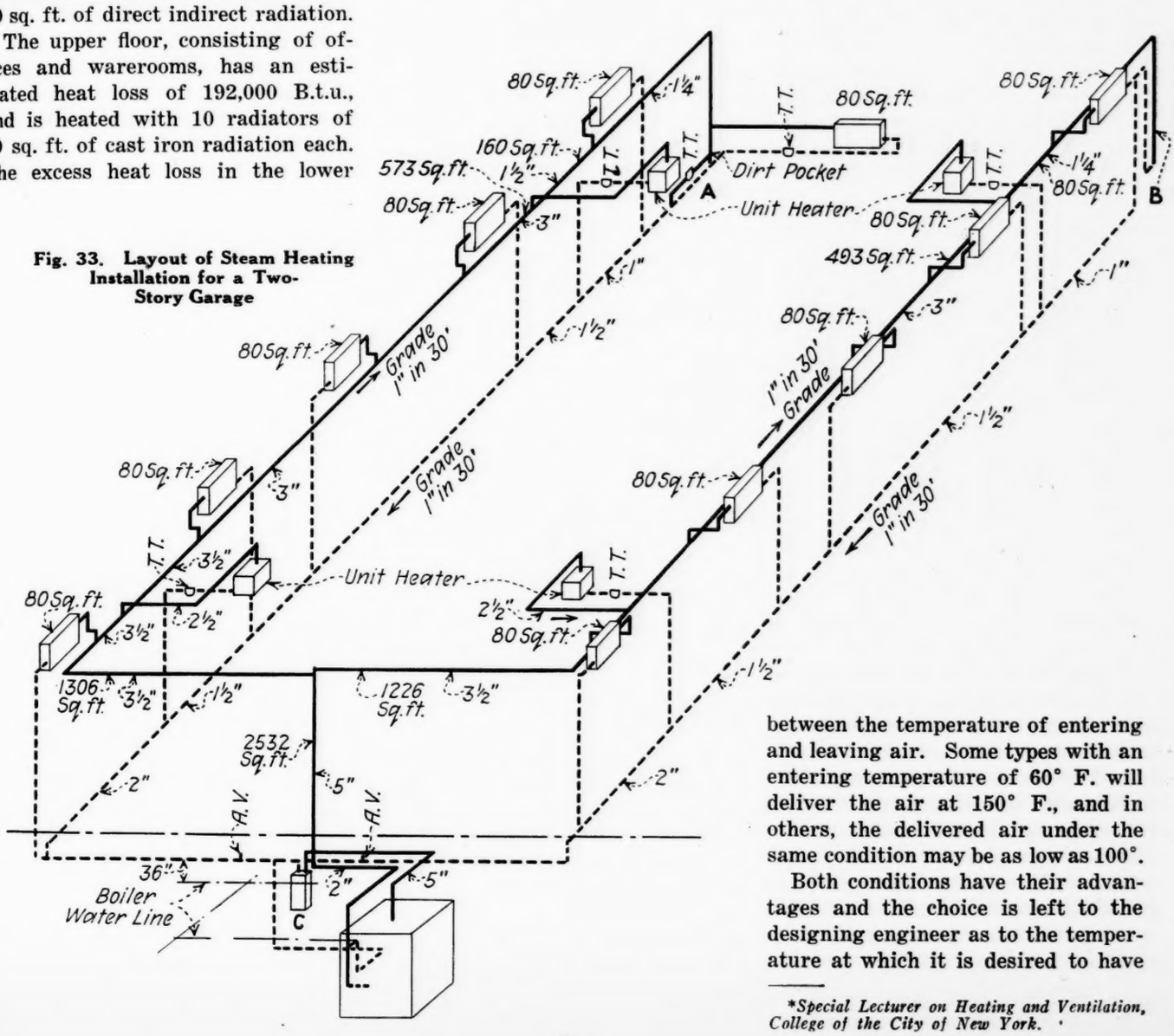


Fig. 33. Layout of Steam Heating Installation for a Two-Story Garage

*Special Lecturer on Heating and Ventilation, College of the City of New York.

the air delivered. The capacities and ratings are given in the catalogs of the various manufacturers. In this case where the temperature in the garage is to be maintained at 65° F. we have selected a unit heater with a rated capacity of 410 sq. ft. of equivalent cast iron radiation, circulating 1800 c.f.m. with an entering air temperature of 60° F., and the temperature leaving the heater at 111° F. For the lower floor the amount of radiation needed is

$$\frac{390{,}000 \text{ B.t.u.}}{240 \text{ B.t.u.}} = 1625 \text{ sq. ft. of cast iron radiation}$$

and it is therefore necessary to use

$$\frac{1625 \text{ sq. ft. of cast iron radiation}}{410 \text{ sq. ft.}} = 4 \text{ units}$$

It would be well for the student at this stage to review some of his studies and check the relationship between the amount of air delivered, its temperature and the equivalent square feet of cast iron radiation. In one of our former installments we found that it required 1/55 or 0.018 B.t.u. to heat 1 cu. ft. of air through 1°, therefore, to heat 1 cu. ft. through a range of 111°—60° = 51° we will require 51° × 0.018 = 0.918 B.t.u. The total B.t.u. needed per hour will then be 0.918 B.t.u. × 1800 c.f.m. × 60 min. = 99,144 B.t.u., which is equivalent to

$$\frac{99{,}144 \text{ B.t.u.}}{240 \text{ B.t.u.}} = 413 \text{ sq. ft. of cast iron radiation.}$$

This method is not exactly precise, as the figure 0.018 B.t.u. is based on 1 cu. ft. of 70° F. air. A closer determination would be by determining the weight of the air and using the specific heat value of 0.24.

The piping must now be designed to carry 413 sq. ft. of cast iron radiation for each heater. The sizes of the mains in a two pipe system are determined similarly to those of the one pipe systems, as described in the January issue, i.e., on a basis of about 1 oz. friction loss per 100 lineal feet of pipe. The risers, however, have a larger capacity due to the fact that the condensation from only the riser itself flows back against the path of the steam. The allowable capacities of the risers are given in Table 17.

The total load on the boiler is 2532 sq. ft. of cast iron radiation.

TABLE 17

MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet per Second	Pressure Drop Ounces per 100 Ft.	Capacity		
			Sq. Ft. Radiation	B.t.u. per Hour	Lb. Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20		40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

From the A.S.H.V.E. Guide, 1931. Copyright American Society of Heating and Ventilating Engineers; reprinted by permission.

Referring to the friction chart, Fig. 32, in the January issue, we find that a 4 in. pipe will carry this amount with a loss of about 1 oz. per 100 ft. This main, however, rises to the first floor ceiling, and referring to Table 17, it is noted that the capacity of a 4 in. riser is but 2042 sq. ft. It will then be best to use a 5 in. pipe from the boiler to the branches at the first floor ceiling. From the friction chart the sizes for the branch mains are readily determined by a moment's inspection.

In all cases where a reduction in the size of the main is made, eccentric fittings must be used. By referring to Table 17 it is found that the risers to the second floor radiators should be of 1¼ in. pipe. It is a safe general rule to make the branches to the risers one size larger than the riser itself when, as in this case, the risers are not dripped. These branches should then be of 1½ in. pipe. This is also true for the branches to the unit heaters. Nearly all supply connections for heaters of this capacity are given as of 2 in.

When the branch connections, as in this case, are taken from the top of the mains and the pipe condensation flowing back against the steam, it is best to make these branches of 2½ in. pipe and reducing to 2 in. at the heater. These branches could have been taken from the side of the main and graded to carry the condensation to and through the heater itself when a 2 in. branch would be ample. The disadvantage is that in case a heater is temporarily shut off, there would be an accumulation of water in the branch which would cause an annoying water hammer, and danger of breaking the connections when the heater is again put in use.

The size of the dry return piping is not quite as important as that of the mains. A restricted pipe would lengthen the warming-up period. A practical rule, much used, is to have the return one size larger than one-half the diameter of the supply. This, however, gives a larger return than is necessary in the larger sizes and holds good only when the supply does not exceed 2½ in. Above this size the return need be only about one-half the diameter of the supply. In a wet return the capacities may readily be doubled.

In the present layout the returns along the floor are still dry returns and a 2-in. pipe is used to carry all the condensation on each branch. Where these two branches unite and drop to the boiler room floor it becomes a wet return riser and a 2-in. pipe is still sufficient.

All boiler heaters should be dripped to return any excess water to the boiler. This drip is often erroneously termed an equalizing pipe, but it becomes one only where a Hartford connection is used. It is usually taken off the horizontal main immediately above the boiler.

In this case the cellar main is graded away from the boiler and the drip connection made at the foot of the main riser.

All radiators and heaters should be equipped with air valves, their locations on the unit heaters and method of installation being generally supplied by the heater manufacturers.

Thermostatic traps are installed on the return from each radiator and each unit heater. These traps are so designed as to pass all air and water of condensation, but close against steam. A similar trap must also be used where the main drops to the return, as shown at point A, or the connection may be made by means of a water seal, as shown at B. Where thermostatic traps are used, a dirt pocket should always be installed. If sand or scale are allowed to reach the thermostatic trap, they may prevent the valve from seating and steam will enter the return mains. If the amount of steam leakage is large, it will find its way through the returns to the other traps, and due to its higher temperature prevent their opening, thus causing the condensation to be retained in all the remaining radiators, causing the boiler to run dry.

With the system in operation, no steam can enter the returns, and it will be noted that the pressure in the returns will at all times be about that of the atmosphere. Any pressure we may carry in the boiler then will cause the water to rise in the return riser sufficiently to overcome this pressure, about 30 in. for 1 lb. per sq. in.

It will be noted that the return lines in the boiler room are but 36 in. above the boiler water line. A pressure, then, of slightly over 1 lb. per sq. in. will cause the water to rise in the return riser to these branches. With a concrete ceiling of 6 in. and a rise in the returns of 4 in. throughout the length of the building, it will also be noted that the remote ends of the returns are but 46 in. above the boiler water line, and a boiler steam pressure of $\frac{46 \text{ in.}}{30 \text{ in.}} = 1.53$ lb. will cause the condensation to be retained throughout the entire length of the run, thus depleting the amount of water in the boiler. Obviously, then, a steam pressure no greater than 1 lb. can be carried in the boiler unless some steps are taken to increase the pressure in the returns when that of the boiler exceeds this amount.

This is accomplished by either the use of a differential loop or a float return trap. A differential loop is shown at point C. A differential loop operating at 6/10 lb. pressure will admit live steam to the returns when the boiler pressure reaches this amount, and there can never be an excess of pressure between the boiler and that of the returns more than 0.6 lb., nor can the height of the water in the return riser exceed 0.6×30 in. $= 18$ in. above the boiler water line. When a flow return trap is used, the height of the water in the return riser above the boiler water line depends on the height at which the trap is located.

New Street Coaches Well Ventilated and Heated

A NEW type of city rail vehicle designed to compete with the private automobile, has made its appearance in the West. The first model built is in successful operation on the lines of the Chicago and Joliet Electric Railway Company, and was described in a recent paper by John R. Blackhall, general manager of the Chicago, North Shore and Milwaukee Railroad Company, presented before the Western Society of Engineers.

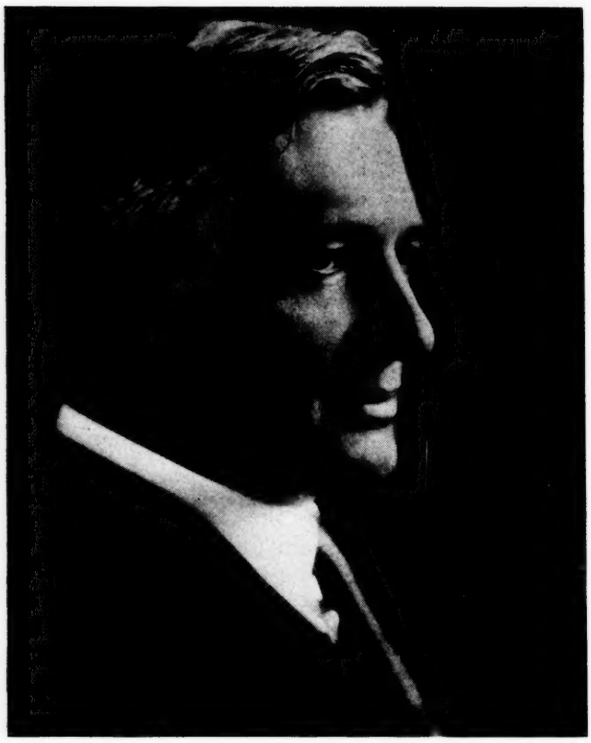
The coach is ventilated positively by a motor-driven volume exhaust blower through grilles in the ceiling and a duct leading to the rear, where the blower is located, and discharging through the opening corresponding in location to the destination sign in the front end. The blower is automatically started when the car stops and exhausts 600 c.f.m. or 40% of the net volume of the car body each minute. Smoke bomb tests showed that the air was about 90% removed in 3½ min. Fresh air enters through louvers in the front near the top, is carried along ducts in the ceiling and down the pier panels where it enters the heater boxes at the floor level. The air is at all times clean and fresh.

The coach is heated electrically from two sources. Seven 700-watt shunt heaters are provided in the heater boxes and are automatically controlled by a thermostat set to cut in and out at 60°. These heaters are used only when the car is still or in the barn before starting out and possibly in the very coldest weather. The principal heating is obtained by utilizing the rheostatic losses in accelerating and the dynamic braking energy during deceleration. From 10 to 20 kw. average power is developed in this way. During the summer time the waste energy is dissipated in resistors exposed to the outside atmosphere; a throw-over switch is provided for directing this energy into the heaters when heat is desired. The heat obtained in this way is ample to provide real warmth for the riders when the run involves a normal number of stops or many grades. It is not intended for long interurban runs with limited number of stops.

Acoustics are also attracting considerable attention in the design of street cars. *The Power Specialist* reports that in a new lot of cars designed for the Cleveland Railways Company, helical gears and pinions were substituted for the older forms between motors and axles to reduce gear hum and clatter, particularly when coasting. In one of the cars acoustical insulation was applied, the ceiling, sides and floors being padded to cut down rattles and the transmission of noise from other parts. The insulation also keeps the car cooler in summer and warmer in winter.

Tests were conducted in the car thus treated, in a similar car, but not so treated, and in a twelve-year old car. The noise intensity measured in decibels was 87.5 decibels for the old car, 72.8 decibels for the new untreated car, and 67.6 decibels for the treated car. This compares favorably with the standard Pullman car, which averages 70 decibels with windows closed.

Willis H. Carrier Elected President of A.S.H.V.E.



Willis H. Carrier *Blank & Stoller*

WILLIS H. CARRIER, the world's outstanding authority on air conditioning, was elected president of the American Society of Heating and Ventilating Engineers at the annual meeting of that organization, held at the William Penn Hotel, Pittsburgh, Pa., January 26-29.

At the Monday afternoon session, January 26, following the report by President L. A. Harding and Secretary A. V. Hutchinson, R. P. Bolton presented his paper telling of "Progress of Heating and Ventilating during the Past Quarter Century." S. R. Lewis was not present and his paper, "Developments in Heating and Ventilating during 1930," was presented by title.

Professors A. C. Willard and A. P. Kratz of the University of Illinois presented a much-discussed paper on "Heat Emission from the Surfaces of Cast Iron and Copper Cylinders Heated with Low Pressure Steam." The authors attempted to show why a cast iron cylinder with walls 0.25 in. thick will transmit 50% more heat than a copper cylinder of the same size with walls 0.01264 in. thick, in spite of the fact that copper has a conductivity of eight times that of cast iron. The overall heat transmission of the iron is better than the copper, according to the authors, because of the greater surface emission of the iron.

At the Tuesday morning session, January 27, the report of the *Guide* Publication Committee was presented. It showed an estimate surplus of $18,500 from the publication of the 1931 *Guide*.

"Development of the Ventilation System of the Holland Tunnel,"* by A. C. Davis, "Noise Reduction in Ventilating Units,"* by Warren Ewald, and "Noise and Ventilation," by G. T. Stanton, were presented at this session. The paper on "Some Studies on the Absorption of Noise in Ventilating Ducts," by G. L. Larson and R. F. Norris was held over. There was considerable discussion on Mr. Ewald's paper.

The preliminary draft of the Proposed Code for Testing and Rating Concealed Gravity Type Radiation caused considerable discussion. The code was prepared by a committee consisting of Reuben N. Trane, chairman, E. H. Beling, W. F. Goodnow, Hugo Hutzel, A. P. Kratz, E. J. Vermere, and O. G. Wendel. The object of the code is to provide a practical method for testing and rating concealed radiators so that the manufacturer can supply the heating engineer with necessary and uniform data for designing systems employing such radiation.

The code, as submitted in preliminary form, provided for test apparatus, procedure, and standard conditions for both steam and water systems. It was finally accepted provisionally with the recommendation that the committee continue its study of the code, and report further at the summer meeting of the society.

**Abstracts of these papers appear on pages 76-79.*

The report of the tellers, submitted at the close of the session, showed that the following officers had been elected to serve during the coming year: President, Willis H. Carrier; first vice-president, Prof. F. B. Rowley; second vice-president, W. T. Jones; treasurer, Frederick D. Mensing. Members elected to the Council were: D. S. Boyden, E. K. Campbell, E. O. Eastwood, Roswell Farnham, and E. Holt Gurney.

At the Tuesday afternoon session, D. R. Brewster presented his paper "Air Conditioning as Applied to Furniture, Fixtures and Other Interior Woodwork."* "Air Conditioning in the Bakery" was presented by W. L. Fleisher. A. H. Candee read and discussed his paper "Air Conditioning for Railway Cars." The Committee on Oil Burning Devices, of which L. E. Seeley is chairman, presented the Proposed Code for Testing Steam Heating Boilers Burning Oil Fuel, the purpose of which is to provide a standard method for conducting and reporting tests to determine heat efficiencies and performance characteristics. The code was tentatively accepted for use and referred back to the committee for further study.

At the Wednesday morning session, January 28, O. P. Hood, chief mechanical engineer of the U. S. Bureau of Mines, addressed the meeting. F. C. Houghten, Carl Gutberlet and C. A. Herbert presented their paper "Air Leakage Through Various Forms of Building Construction," and Mr. Houghten and Mr. Gutberlet also presented a paper "Capacity of Return Mains for Steam and Vapor Heating Systems." The paper by F. G. Outcalt, "Welded Piping for Building Heating Systems," aroused considerable discussion, and the importance of skilled welding operators was stressed, as well as the necessity for careful specifications. H. B. Meller presented a paper "Air Pollution from the Engineers' Standpoint," and M. D. Engle one on "Smoke and Dust Abatement." F. C. Houghten presented the report of the Research Director.

At the Thursday morning session, the following papers were presented: "Spectral Distribution of the Energy Radiated from Metallic Surfaces at High Temper-

atures,"* by G. R. Greenslade; "Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces,"* by L. W. Schad, and "Wind Velocities and Temperature Gradient Near Surfaces," by F. C. Houghten and Paul McDermott.

At the Thursday afternoon session, papers were presented by Ben Gröndal on "Utilization of Hot Springs for Heating in Iceland," and Elliott Harrington on "Off-Peak System of Electric Heating."* Perry West's paper on "Heating with Exhaust Steam,"* was also presented.

Following the installation of officers, it was announced that the F. Paul Anderson Medal for 1932 is to go to a member of the Society for "outstanding achievement in heating and ventilating." Further announcement will be made in the Journal.

The following council committees for 1931 were appointed: Executive Committee, F. B. Rowley, chairman, D. S. Boyden, J. D. Cassell; Membership Committee, Roswell Farnham, chairman, R. H. Carpenter, E. B. Langenberg; Finance Committee, F. C. McIntosh, chairman, John Howatt, W. T. Jones; Program Committee, G. L. Larson, chairman, J. J. Aeberly, L. A. Harding; Publication Committee, W. A. Rowe, chairman, E. O. Eastwood, F. D. Mensing; Increase of Membership Committee, C. W. Farrar, chairman, E. K. Campbell, E. H. Gurney.

About 630 persons attended the meeting.

*Abstracts of these papers appear on pages 76-79.

Use of Copper Tubing between Oil Tank and Burner

A RECENT development in oil burner installation methods to which comparatively little thought has apparently been given is the use of copper tubing in place of the rigid pipe now used between the oil supply tank and the burner. Perhaps the primary reason for the delay in the widespread use of this improvement in modern pipe-fitting practice may be due to the lack of a proper presentation of this method to the oil heating industry. In order to carry out this endeavor we recently sent a questionnaire on this subject to our entire membership. An analysis of the replies together with other essential information we have compiled is given in the following article:

Half of the companies never used copper tubing and they were equally divided between manufacturers and dealers. A number of them expressed considerable interest and desire for further information.

Practically all of those companies who had used copper tubing secured very favorable results and endorsed its use. Their cost, taking labor and material into consideration, was no greater than that for iron pipe. They found it easier to handle and improved the appearance of an installation. The size generally used was ½ in. tubing and installed on gas lines as well as suction and return lines from the tank to the burner.

One dealer with considerable and very favorable experience on copper tubing cautioned against locating it near the surface of loose soil in a driveway, where heavy trucks may pass over. Any difficulty in a situation of this kind will be eliminated by placing the tubing deep enough or using an old 1 in. pipe as a sleeve. The following report of a test made by one of the prominent manufacturers of copper tubing is a good indication of its ability to withstand all normal external pressures without flattening: "A load of one ton was concentrated on a piece of ¾ in. copper tubing about 1 in. long and produced an approximate distortion of 75% of the diameter of the tube."

The following are some of the outstanding advantages of copper tubing:

1. The tubing being flexible, long runs can be made without the use of fittings and most changes in direction are made by bending the tube. In other words, one piece of copper tubing can be run from the bottom of the tank to the burner, if desired, without fittings of any kind, with the exception of means for anchoring. The reduction in the number of fittings practically eliminates the possibility of leakage which occasionally occurs due to poorly cut threads or improperly assembled joints on ordinary iron pipe. Everyone undoubtedly appreciates the seriousness of leakage occurring in lines underground. It involves not only locating and repairing the leak itself but also removing earth or cement. This is not only an expense to the dealer but aggravating to the owner. The flexibility of tubing is of great value where the ground or tank settles appreciably.

2. The possibility of scale to work loose and clog nozzles, passages or strainers is practically eliminated, thereby contributing to more reliable operation of the burner.

3. The smooth interior of the tubing and the elimination of elbows and other fittings assist in producing a free flow of oil. In every case where a fitting of some kind is eliminated a smooth bend of considerably greater radius is substituted and assists in reducing the load on the oil pump. The pump is able to operate more effectively as a result of this reduction in friction.

4. Practically unaffected by corrosive substances ordinarily existing in the soil. This helps to assure permanence of piping.

5. While the tubing may cost more per foot than iron pipe, the fewer fittings required cost less than the total number of threaded fittings for the same job and together with the economy in labor offset this difference. On an average domestic installation the difference in cost of tubing and fittings compared to iron pipe and fittings is somewhere between $2.50 and $3.00.

6. The flexibility of tubing assists in eliminating noises due to vibration of the burner and may be "telegraphed" through the house.

In using copper tubing, joints can easily be made with the aid of a few simple tools. The tools required are a hack-saw, a flanging tool, a hammer and a wrench. The methods to be followed are equally simple and are:

Use a hack-saw with fine teeth and cut the tube off square.

Ream or cut two good shavings from inside of tube with a jack knife which will make a bevel that will accommodate itself to the face of the tool and facilitate flaring.

Slip the sleeve nut member of the fitting over the tube as this cannot be done after the tubing has been flared.

Oil face of tool, insert in tube and hold in position with the thumb and first finger.

Use a light hammer and strike the center face of the tool, turning slightly after each blow. The loose nut on the tool acts as a die to prevent overflaring and a change in the "feel" of the flow will be noticed when the proper diameter has been reached.

Withdraw the tool, connect the two members of the fitting and draw tight, then back off the sleeve nut a half or full turn and tighten again. This secondary compression reams the male member of the fitting into the flare of the tube, making a seat that will be permanently tight.

Copper tubing may be procured in 20, 45 or 60 ft. coils and in ⅜ in., ½ in., ¾ in. or larger sizes. It is recommended by the National Board of Fire Underwriters and its increased use will benefit the entire oil heating industry.—*Bulletin, American Oil Burner Association.*

Abstracts . . .

Reviews and Summaries of Current Papers, Books and Pamphlets

Mechanism of Heat Flow in Fibrous Materials

by **S. L. Finck**

Bureau of Standards Journal of Research, United States Department of Commerce, Vol. V, No. 5, November, 1930, pages 973 to 984

A SYSTEMATIC investigation of the thermal conductivities of specimens composed of fibrous materials has been made. The experiments were designed to show the effects of such factors as density of packing, arrangement of fibers, kind and size of fibers, moisture content, air convection, and radiation, on the resultant conductivity of the specimens. Data on various mixtures of fibers were obtained, and it was found that in most cases the conductivity of the mixture lies between those of the constituents, taking the conductivity of each constituent which corresponds to a density equal to the density of the mixture.

It was found that for a given fibrous material and a given density, the conductivity may vary by several hundred per cent, depending entirely on the arrangement of fibers. The maximum conductivity is obtained when the fibers are parallel to and the minimum conductivity when the fibers are perpendicular to the direction of heat flow. There is included a discussion of the part which contact resistance between the fibers may play in the over-all heat transfer.

At very low densities it was found that radiation plays an important part in the heat transfer; air convection plays a very minor part.

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Heat-Flow Meters and Thermal Conductivity Measurements

by **F. G. Hechler** *and* **E. R. Queer**

The Pennsylvania State College Bulletin, Engineering Experiment Station Series, Bulletin No. 37, August 22, 1930. 60 pages, 30 figures, 9 tables; 6 in. x 9 in. 50 cents.

DESCRIBES the construction and methods of calibrating four corkboard heat-flow meters approximately 3 ft. x 3 ft. in size. Data resulting from calibration tests for two meters made of ¼ in. transite, and for six made of ⅛ in. ebony asbestos board, are presented. Includes the numerical results obtained from measuring the heat flow through dwelling-house walls and ceilings, through brick walls and concrete roofs, including the effect of spraying water on a brick wall, through a glass window, and through the walls of a concrete storage cellar. Contains four appendix sections of twelve pages.

Possibly the most interesting feature of the bulletin is the heat flow time charts which are included. The change in the amount of heat flowing at various times is clearly shown. The effect of the time lag due to the mass of the materials is shown in the case of a concrete roof and a brick wall 18 in. thick. The flow of heat through the lighter roof section is much more quickly sensitive to temperature than is the massive wall.

Data showing the effect of wetting a wall on the heat transmitted are unusually interesting. The authors make the point that building walls may often be damp during a considerable portion of the heating season, and point out that in such case the heat transmitted may be considerably greater than would be found if the wall were thoroughly dry.

Some of the figures found by these tests are compared with those obtained by the use of formulae and laboratory constants for computing heat transmittance. In some cases, it appears that substantially the same result is obtained by either method.

In their introductory statement, the authors state: "Heat-flow meters are useful because they furnish a convenient method for measuring the heat flow through the walls of buildings under service conditions . . . . The thermal resistance of most buildings and insulating materials is known from laboratory tests made under ideal conditions with dry samples, but it is equally important to know whether the transmission for a compound wall, when calculated from these values, agrees with the actual flow in service. Heat-flow meters give valuable data on this, and results obtained with them confirm the reliability of the constants used and the methods of calculation employed."

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Air Conditioning for Railway Passenger Cars

by **A. H. Candee**

A.S.H.V.E. paper, annual meeting January, 1931. About 4000 words, 14 photos, 3 tables.

AUTHOR estimates, on the basis of present named trains, that there is a potential field of 15,000 passenger cars which may eventually be air conditioned if the use of this method reaches such a point that it is economically justified. He regards the two air conditioned diners now in operation—one owned by the Baltimore and Ohio and one by the Santa Fe—as being trial installations. The 14 photos show interior and exterior views and some of the apparatus features of these two cars.

It appears that the two cars are demonstrating the feasibility of applying refrigerating apparatus for cooling but that the conditioning apparatus as now installed will very likely be greatly changed if large numbers of cars are to be conditioned. Attention is called to the fact that in neither of these cars is the winter heating accomplished by means of the air circulation apparatus installed, nor is humidity closely controlled.

The author concludes that any air circulating system for cars should include one or more air filters equipped with manually-operated shutters which may be adjusted at will, a circulating fan and a duct system for distributing the air. He feels that the usual figures of 15 c.f.m. per passenger may eventually be reduced somewhat.

Available means of securing cooling are reviewed and while mechanical apparatus is used on the two cases, the others have possibilities which may be developed. He also discusses the six means of providing energy for driving a mechanical refrigerating apparatus and feels that the axle-driven generators used on the present cars may be superseded.

The heat load on a diner or sleeper is computed to be from 60,000 to 80,000 B.t.u. per hr.

The problems to be solved include the matters of the best method of refrigerating, of insulating the cars, of supplying the power and of arranging the equipment. Before the use of these cars can become general, the railroads must train a widely-scattered organization to keep them operating.

Noise Reduction in Ventilating Units

by **Warren Ewald**

A.S.H.V.E. paper, annual meeting January, 1931. About 2000 words. No illustrations.

A DESCRIPTIVE paper. Lists some of the sources of noise in a unit ventilator. Discusses these items and their prevention at the source. Describes briefly some tests made to study the effectiveness of certain sound-absorbing materials placed inside the cabinet. Discusses briefly interference and resonance in the noise from unit ventilators. Calls attention to the importance of the noise problem in unit ventilator development.

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Air Conditioning as Applied to Furniture, Fixtures and other Interior Woodwork

by **D. R. Brewster**

A.S.H.V.E. paper, annual meeting January, 1931. About 3000 words, 3 tables, no illustrations.

THE change in moisture content in wood exposed to the air is directly related to the temperature and humidity conditions of the air since wood is hygroscopic.

According to the author, "the atmospheric changes in buildings are ordinarily so gradual from season to season that distortion in furniture or woodwork due to fiber strains seldom occurs under the usual conditions of use." But in spite of the design allowances made for shrinking and swelling, the use of some artificial humidification may be helpful in preventing damage to furniture and other woodwork.

Tabulated results of some recent studies to determine the seasonal changes in the moisture content of unfinished wood in several cities are presented. The author feels, however, that at certain more or less unusual and abnormal times these figures may be exceeded.

Surface coverings, such as paint and varnish, are very effective in preventing excessive moisture changes in wood, but much depends on the thoroughness with which they are applied. Several examples are cited where valuable antiques, museum pieces and imported furniture have been ruined because of excessively dry atmospheric conditions. The importance of air conditioning in a museum is stressed.

Data of value in preparing lumber for use in the market are presented, and the author states that kiln-drying and controlled storage are desirable practices.

In order to avoid the difficulty of surface condensation which results in cold weather when high humidities are artificially maintained, the author suggests that the high humidities be maintained during the relatively longer periods of comparatively mild weather which make up a considerable part of the heating season. This would put the furniture and woodwork in better condition to pass through the cold weather period with a lower relative humidity maintained.

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Off-Peak System of Electric Heating for Buildings

by **Elliott Harrington**

A.S.H.V.E. paper, annual meeting January, 1931. About 2500 words, 5 photos, 1 drawing, 1 chart, 1 table.

DISCUSSES the idea of off-peak energy and explains why the unit rate is low. Data from seventeen houses heated by an off-peak storage method in Fort Wayne, Indiana, are presented in tabular form. The installed kw. in these houses varies from 80 kw. in a house of 44,840 cu. ft. to 30 kw. in a well insulated house of 9130 cu. ft. With off-peak power at two cents per kw-hr. for the first 400 kw-hr. per month and one cent per kw-hr. thereafter, the cost of heating the 9130 cu. ft. house was $190, and of the 44,840 cu. ft. house was $1300.

The author lists six advantages of this form of heating. A sketch and three photos together with a word description show the principal features and the operation of this system of heating. Domestic service water can be heated electrically by this method.

About fifty installations of electric heating have been made in the past three years. The author sees a considerable use for this method in the future, mentioning automatic telephone exchanges, residences, apartment buildings and railway service buildings as being good prospects.

He states that under present conditions many electric utilities have available enough off-peak energy to heat 2% to 5% of the homes in their territory. He feels that future load developments are likely to increase the amount of off-peak energy available.

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Welded Piping for Building Heating Systems

by **F. G. Outcalt**

A.S.H.V.E. paper, annual meeting January, 1931. About 2000 words, 4 sketches and 7 photos.

EFFICIENCY of welded joint is defined. Factors entering into the economy of welded construction briefly summarized. Dimensional data for proportioning V butt welds given. Describes and illustrates design of weld at fittings, branches, and crosses where these are fabricated from standard pipe. Mentions use of forged fittings and use of forged seamless short radius quarter and 180° elbows. Gives example of weight saving accomplished in a certain header using welded construction instead of cast-iron flanged fittings. Adaptability of welded construction to tight working space illustrated by an example which describes how a main and branches were installed in a 14¾ in. space when the largest pipe was 12 in. diameter. Mentions use of loops for providing for expansion. Emphasizes the importance of securing good procedure control during erection if successful results are to be obtained.

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Efficiency and Economy of Underground Steam Distribution System

by **W. R. Woolrich** *and* **Leo Holdredge**

Bulletin No. 10, University of Tennessee Engineering Experiment Station, November 1930, 35 pages, 6 x 9, 15 figures.

FOLLOWING a list of seven items called "Requirements of a Successful Underground Distribution System," which items include support, drainage, insulation and expansion, ten kinds of underground construction in use are listed, and described and discussed separately.

The "insulation efficiency" is defined as:

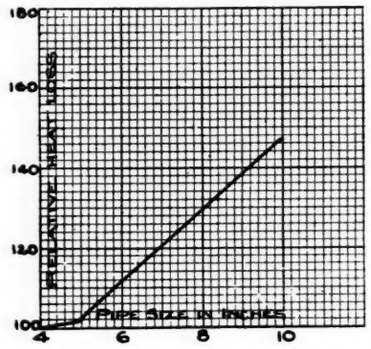
$$E = \frac{B - C}{B}$$

when E = Efficiency of covering
B = Heat loss from bare pipe in still air
C = Heat loss from covered pipe.

The authors state that the pipe covering which gives the best efficiency by the above method of calculation may not be the best covering, for they work out two problems showing that the total heat loss from a given size and length of pipe depends on the pressure as well as on the insulation efficiency.

Elements which influence the efficiency and economy are listed as: Steam pressure, pipe size, thickness of covering, depth of burial, conductivity of covering, kind of soil, moisture content of ground, workmanship in construction and air circulation. Each is separately discussed.

Under the heading of "Design for Best Economy" the following example appears:



The Assumed Data for this Problem are as follows:

Length of Steam Main, 5000 ft.
Maximum amount of steam needed, 6000 lb. per hr.
Average Demand, 40% of maximum.
Boiler Pressure, 100 lb. per sq. in.
Required Pressure at End of Main, 25 lb.
Ground Temperature, 60°.

What is the Most Desirable Size Pipe to Install?

From Calculation, a 4-inch pipe is the smallest that will deliver the maximum amount of steam at the end of the line at 25 pounds pressure. The curve shows the relative heat loss from larger sizes of Pipe when the Average Load (40% of the maximum) is on the line. Note that the Heat Loss Increases very little up to a 5-inch pipe, but there is a rapid increase when a pipe larger than 5 inches is used.

The section concludes with: "In general it may be said that the smallest pipe which will deliver the maximum amount of steam needed at the required pressure is the one that will show both the best thermal and commercial efficiency."

Operation of the line should be accomplished using the lowest pressure possible if the best economy is to be obtained. Presumably this applies to the use of saturated steam, for the figure presented to support this statement applies only to such steam.

It should be noted that the author states: "All the above refers to the transmission of live steam. The use of exhaust steam is a separate problem."

Tables are presented showing the results obtained by the authors in four tests made during July, 1930, on a section of underground steam line. The test apparently consisted of keeping the test section filled with steam of a known pressure and quality for a period of four to five hours. Presumably the steam in the section under test was stationary since the section was "isolated." The "insulation efficiency" was calculated for each test and was found to increase from 87.2% with 10 lb. (gauge) steam to 90.2% with 98 lb. steam. The plotted results show the economy based on the calculated heat loss to be nearly a straight line function of the pressure change.

As a result of temperatures measured inside the conduit, the authors conclude that a vacuum return line should be carried in the same conduit as the steam line, and point out that the losses from such a line are not great.

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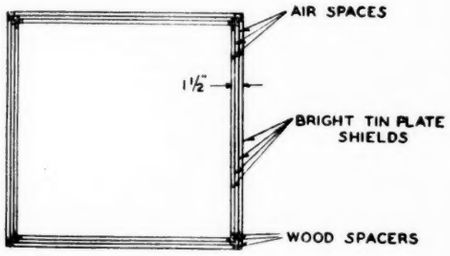
Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces

by **L. W. Schad**

A.S.H.V.E. paper, annual meeting January, 1931.
About 4000 words, 5 figures.

APPARATUS is described for testing the heat transfer from surface to surface of three parallel layers of bright tin separated by one-half inch air spaces. A box 27½ in. × 27½ in. × 30 in. was used to measure the transfer and to thus determine the coefficient. The surface temperatures were measured by thermal couples. The figure shows a plan view of the box used.



Test Box Used in Tests by L. W. Schad

Solution of the usual conduction formula using the test results gave the following values:

t_1	t_2	k_t
69°	50°	0.24
98°	78°	0.30
112°	83°	0.32
165°	92°	0.35

In view of the test conditions no great accuracy is claimed for these results.

In addition to the above method the author also set up an apparatus for testing the best transfer by the guarded hot-plate method. As before, three parallel layers of bright tin separated by one-half inch air spaces were used as a test specimen. The specimen, like the box walls, was of 1½ in. thickness. The tests results are shown as follows:

t_1	t_2	k_t
176°	103°	0.40

The author feels that this result is perhaps slightly too high.

The author then shows the mathematical analysis to develop equations suitable for computing the value of the heat transfer coefficient (surface to surface) based on the fundamental physical equations commonly used for radiation, convection and conduction. He develops such equations for use both where an air space is not screened and where any number of successive screens are introduced into the space.

Assuming numerical values and substituting them into the equations he shows that for the three following cases the results computed are:

Case 1. 1½ in. air space without screens bounded by two parallel surfaces such as cork, paper, wood, etc..... $k_t = 2.66$

Case 2. 1½ in. air space without screens bounded by two parallel surfaces such as bright metals, $k_t = 1.11$

Case 3. 1½ in. air space like Case 2 but with two bright tin plates introduced which divide the 1½ in. air space into three ½ in. spaces..... $k_t = 0.44$

(Hot surface temperature 212° and cold surface temperature 32° in all three cases).

The amount of heat transferred in each case by radiation, by conduction, and by convection is also shown.

The results in Case 3, which is essentially the same case as was used for the direct tests, is thus 0.44 by computation as against about 0.38 by test. In view of the nature of the computation the results by the two methods may be considered in good agreement.

The effect of introducing the bright metallic screens into the air space is to increase the transfer by conduction in an amount depending upon the extent to which the air path is displaced by metal. The heat transfer by convection is decreased as follows:

$q = K_s (t_1 - t_2) d$ without screens, then with n screens

$$q = \frac{K_s (t_1 - t_2) d}{(n + 1) d}$$

The heat transferred by radiation is decreased as follows: If q = transfer by radiation without screens, then

$\frac{q}{n+1}$ = the transfer by radiation with n screens.

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Noise and Ventilation

by **C. T. Stanton**

A.S.H.V.E. paper, annual meeting January, 1931. *About* 3000 *words,* 4 *figures. Bibliography of six titles.*

AUTHOR classifies the problem of reducing noise as follows: (1) Reduction of noise produced at the source; (2) Reduction of transmission of noise from the source to those areas under consideration; (3) Acoustical treatment in the area under consideration, to reduce the effect of noise.

He states that the noise problem in ventilation apparatus falls under the two heads (1) and (2).

Describes and discusses the available methods of measuring and evaluating noise. Discusses and lists some of the variables in the evaluation process, and outlines a shop method of determining certain of the noise characteristics of apparatus. Explains his method using a blower as an example.

Author states that by using some standard test procedure the following benefits, which he believes timely and necessary, are obtainable.

(1) Shop measurements may be converted into performance ratings in any given location; (2) A basis of rating equipment on noise performance may be established; (3) Standards of minimum allowable noise for different types of systems, relative to air delivery, etc., may be established: (4) All data taken will be capable of correlation and comparison, permitting more rapid accumulation of empirical data.

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Special Distribution of the Energy Radiated from Metallic Surfaces at High Temperatures

by **G. R. Greenslade**

A.S.H.V.E. paper, annual meeting January, 1931. *About* 4000 *words,* 12 *figures.*

DESCRIBES the apparatus and procedure used in laboratory tests by the author in connection with the following metals: iron, tungsten, molybdenum, nickel, cobalt, platinum, iridium and rhodium. Curves are presented showing the resulting data. The curves are in scientific units and appear to have no present application value to the engineer. The apparatus is extremely ingenious and the author's account of the care used and precautions taken indicate that the data were arrived at in accordance with the very best laboratory and scientific precision obtainable.

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Utilization of Waste Heat

by **Perry West**

A.S.H.V.E. paper, annual meeting January, 1931. *About* 4500 *words,* 8 *figures,* 3 *tables.*

LISTS and discuses some of the sources of waste heat which may be used for heating and ventilating, and points out that the use of the waste must be economically justified.

Takes up the commonest case of waste heat utilization viz., where exhaust steam is used for heating. Tabulates and discusses the causes of fluctuations in the quantity of exhaust available for heating, and in the heating load. The several combinations of apparatus and operation in use for adjusting and coordinating the fluctuations are briefly described.

A daily load curve for a hotel in the cold months is presented as is a similar one for the hot months of the year. Like curves are shown for an office building and for a high school. The author states that such curves must be computed for each hour of the day and for the various days of the year but feels that typical curves for each of the four seasons of the year are generally sufficient. He gives figures for use in estimating the annual heating demand and for reducing the quantities to a monthly basis.

With the heating load and its fluctuations thus determined, attention is turned to methods of determining the amount of exhaust available. In order to do this the water rate of the engine or turbine at various loads must be known. Charts are presented for estimating these amounts and methods of determining the condition of the exhaust steam are illustrated.

A somewhat similar analysis of the relations between the fluctuations of the demand for process steam and for steam for electricity in the load in a certain industrial plant is presented. In the course of this analysis an interesting application of the Willans line is illustrated. As a result of his analysis the author feels that this particular plant would be justified in building its own power plant to furnish both power and process steam.

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Some Studies on the Absorption of Noise in Ventilating Ducts

by **G. L. Larson** *and* **R. F. Norris**

A.S.H.V.E. paper, annual meeting January, 1931. *About* 3000 *words,* 4 *photos,* 5 *charts.*

A DISCUSSION of the sources of noise in a fan-and-duct arrangement, and of their transmission, together with a short statement of some of the principles of sound measurements.

The paper is devoted principally to a description of the procedure and the results of some laboratory tests made to learn the effect on sound intensity of several arrangements of acoustically-lined and unlined ventilating ducts, and the effect of such linings on the resistance to air flow at velocities ranging from 500 to 5000 ft. per min.

A test duct 10 in. x 10 in. inside and 30 ft. long was built of 2 ft. sections. It was carefully sound-insulated to make sure that the noise coming through was really from the duct. The two-foot sections were both insulated and uninsulated. By the arrangement of the duct it was possible to get arrangements varying from an all-lined duct to an entirely-unlined duct.

Measurements of the noise were made by the use of a microphone and acoustimeter.

The tests showed that the duct noises can be greatly reduced by lining them with sound insulating material. The best results were obtained when the ducts were lined throughout their length. The tests brought out the fact that when the lining material was placed along 6 lin. ft. of the duct near the inlet it was more effective than when placed along 18 lin. ft. near the outlet. The authors also conclude that the tests showed that placing acoustical material in the ducts as a lining makes it possible to use higher velocities than are in general use without resulting in disagreeable noise.

• • •

Development of the Ventilation System of the Holland Tunnel

by **A. C. Davis**

A.S.H.V.E. paper, annual meeting January, 1931. *About* 4000 *words,* 5 *sketches,* 9 *photos.*

A DESCRIPTIVE account of the research work preliminary to the development of the ventilation system, followed by a short description of the principal ventilation apparatus features of the completed tunnel.

Correspondence

Characteristics of Unit Heaters

EDITOR HEATING AND VENTILATING:

After carefully reading J. H. Smith's very interesting and instructive article entitled "Design, Operating Characteristics and Selection of Unit Heaters," appearing in the December HEATING AND VENTILATING, some questions come to mind which the writer believes should be discussed. The two fundamental laws mentioned by Mr. Smith—one, heated air rises if uncontrolled; two, air is most easily moved horizontally within its own temperature level—while true, are frequently misapplied in discussing unit heaters.

It is undoubtedly true that heated air rises if it is not forced downward into colder strata, as this action naturally follows from a fundamental consideration of buoyancy. It is for this reason that many engineers have found that the most effective heating system is to deliver the heated air close to the floor with a fairly high horizontal velocity so that the heated air definitely warms the lower strata before it has a chance to rise upwards above the breathing zone. It is also true that unit heater manufacturers have paid an increasing amount of attention to the desirability of deflecting heated air downward whether the heater be of the floor mounted type or suspended type.

In ordinary heating practice with the heated air entering a building at a temperature in excess of 120°, the buoyancy effect tending to cause the heated air to rise is so much greater than any pull down effect, due to floor recirculation, that it may be said that the circulation of heated air downward through the breathing zone is accomplished by blowing the air in a downward direction and not due to the effects of floor recirculation.

The second law mentioned above has practically no bearing on unit heater practice. If we assume that air leaves a unit heater at a temperature of 120° in a horizontal direction and is immediately reduced to, say 90° or 100°, if the heater outlet is at an elevation of 7 or 8 ft., it is obvious that no commercial building would be heated to 90° or 100° over a complete horizontal plane at any such elevation from the floor. Therefore, warm air leaving a unit heater cannot move horizontally in its own temperature level and must be forced downward against the effect of buoyancy in the case of any ordinary unit heater installation.

In order to control the discharge air it not only is desirable to have a relatively low final temperature but it is indispensable to have a downward deflection if the heater discharge is located above the breathing zone. Therefore, there must be a proper relation between outlet temperature, outlet velocity, angle of deflection and elevation of the outlet in order to secure satisfactory results.

A further consideration of floor mounted units leads one to the conclusion that the fan action of the heater tends to produce a sub-normal pressure at the fan intake. Since a very large part of infiltration undoubtedly results from cracks under doors and opening of doors, etc., it would naturally follow that floor mounted unit heaters would tend to increase the infiltration of cold air, thus offsetting to a considerable degree, the advantages assumed for this type of heater as to effective capacity. In other words, while it is true that greater capacity of heating equipment is required when the inlet temperature is raised by elevating the heater from the floor, the additional infiltration caused by floor mounting of itself requires additional capacity tending to reduce the difference between the two methods of heating.

C. A. CARPENTER.

Pittsburgh, Pa.

Ventilation Laws

EDITOR HEATING AND VENTILATING:

I find myself generally in agreement with your editorial in the September issue on Ventilation Laws.

I do not know of a State where the statutes are in compliance with the best current practice for ventilation of schools and public buildings. It is high time for agitation to commence in the more progressive States to have these laws revised, and it has been found very useful to have a definite program in the form of a suggested uniform law which can be adopted. The American Society of Heating and Ventilating Engineers should appoint a committee to prepare and submit for approval by its membership such a law, and its adoption by a few States would be quickly followed by the rest.

C. A. BOOTH

Buffalo, N. Y.

Injector Used for Creating Vacuum

EDITOR HEATING AND VENTILATING:

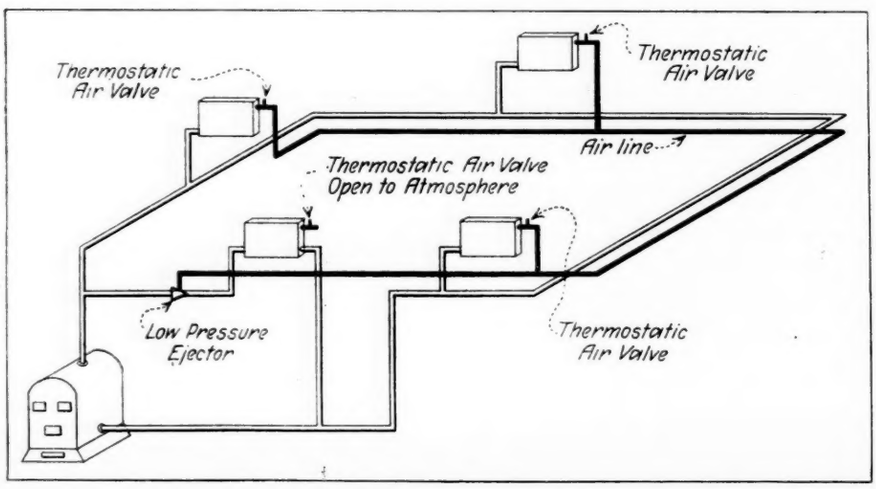
Can you supply me with names of manufacturers fabricating a very small vacuum pump which could be used in a garage for following a vacuum at that point on a gas burner job which does not carry sufficient pressure to force the steam to the garage? If a vacuum could be maintained at that point steam would follow.

J. A. A.

Racine, Wis.

We believe that the best solution in a small installation of this nature is the use of a small injector working on low pressure to create a vacuum in and through an air line. Refer to the sketch below showing the layout and method of installation. You will notice that the injector draws the air from and creates a vacuum in all the radiators except the last, which is furnished with the steam from the injector itself and should be fitted on the two-pipe principle.

Piping can be arranged to suit the building. The steam used in the injector and the last radiator will take care of 15 times as much as is used in the other radiators. That is, the last radiator should be large enough to take care of 1/16 of the entire radiation.



Layout of System Using Injector for Creating Vacuum

Correspondence

Warm-Air Heating and the Degree Day

EDITOR HEATING AND VENTILATING:

Your "Degree Day Handbook" has proved helpful, but I would like to suggest an addition to it—a chart showing fuel consumption per square inch of heating pipe for warm-air systems. I have been wondering if it could be done, inasmuch as there is a difference of velocities caused by different heights of registers from castings.

J. B. W.

Utica, N. Y.

The following data have been prepared giving the quantities of the three fuels per square inch of leader pipe per degree day, assuming that the register temperatures are 175°. This temperature is that upon which the Standard Code is based, and calls for a heat emission of 111 B.t.u. per sq. in. of first floor leader, 167 B.t.u. per sq. in. of second floor leader, and 200 B.t.u. per sq. in. of third floor leader. It is based, as are the other degree day data, on the fact that the system is operating at 100% efficiency, a designed temperature of 0–70°, a heating value of 140,000 B.t.u. per gal. of oil, 12,000 B.t.u. per lb. of coal, and 1000 B.t.u. per cu. ft. of gas. For any other conditions corrections must be made as outlined in the "Degree Day Handbook," and in the data accompanying the monthly degree day charts in HEATING AND VENTILATING.

	Lb. of Coal per Deg. Day per Sq. In.	Cu. Ft. of Gas per Deg. Day per Sq. In.	Gal. Oil per Deg. Day per Sq. In.
First Floor Leader	0.0037	0.044	0.000318
Second Floor Leader	0.0056	0.067	0.000477
Third Floor Leader	0.0067	0.080	0.000571

Heating the Montgomery Ward Building

EDITOR HEATING AND VENTILATING:

Referring to the editor's notes on the article "Heating Economies in Montgomery Ward Buildings," which appeared on page 96 in the November issue of HEATING AND VENTILATING:

Regarding Question 1, it is obvious that a vacuum return line system cannot be maintained at 70° F. inside temperature without intermittent operation, accomplished either by manual control or some sort of automatic temperature regulation.

Question 2 is more involved and really has no bearing on Question 1, as it states "Is it correct to make comparisons on the degree day basis for inside temperatures other than 70°?"

Assuming the standard for inside temperature to be 70° as representing the comfortable temperature for the majority of people, and the standard for outside temperature having been set at 65° F. above which no heat is required inside buildings, as a basis, when heat is required it is desirable to maintain same as near 70° as possible. In applying the degree day basis in calculating the comparison of one system against another, the conditions inside the building should be as near normal as possible, which was the case in this test. The degree day only applies to outside temperatures and is only a gauge by which to make comparisons.

I fail to see any justification in correcting the results indicated in fuel saving by calculating on the basis of the number of degrees difference between the inside and the outside temperatures, which would be penalizing the differential system for making it possible to maintain 70°.

Further, as indicated by the test the outside temperature was 2.17° lower during the differential period of operation than during the vacuum return line period with 4.34 degree days more to compensate for and yet showing a lower fuel consumption. Taking the fuel saving alone, which represents 30% in favor of the differential system, is proof enough without any reference to the degree day method of comparison.

It appears to me that the figures given in the test showing 32.24% saving arrived at by using the degree day method are correct.

JOHN D. SMALL

Chicago, Ill.

EDITOR HEATING AND VENTILATING:

I have just read in the Correspondence columns of your January number several comments on the article "Heating Economies in Montgomery Ward Buildings," which appeared in the November issue of HEATING AND VENTILATING.

With reference to the Editor's note in the earlier issue, it seems that your contention for the 27% saving is based solely on the inside temperature of 73.8°.

You are assuming that this temperature is constant—always 3.8° above the desired temperature. Of course, such is not the case. With the thermostat set 70°, the temperature chart for the return line system probably looked like the Sierra Nevada mountains with high points and low points 15° to 20° apart, that variation being no more than a natural characteristic of the vacuum return line system. The divergence from the desired temperature of 70° is one of its defects.

To try to minimize this fault would be foolish. For the interests of *heating progress*, as well as a matter of logic, it should be rigidly observed in any calculation which involves a comparison with better systems, viz: the differential vacuum system.

On account of the irritating errors in temperature found in the return line system, to my mind the saving of 32.24% is more than a minimum indication of the superiority of the differential system.

Thanks for your heating demand map on page 81 of the January issue. It puts valuable facts in a form easily visualized. Every firm in this field doing a national business should have a copy of it. Keep it up!

C. R. BROWN

Chicago, Ill.

Ventilation Laws

EDITOR HEATING AND VENTILATING:

Relative to the editorial in the September issue of HEATING AND VENTILATING on Ventilation Laws, my own preference is to keep out of such controversial arguments, but even the timid may at times be persuaded to throw off their coat and enter the fray where circumstances justify.

You say that present laws restrict those in charge of building construction to practices which are out of harmony with present-day scientific conclusions on the subject, and yet in the following sentence you admit that all may not be unanimously of the same opinion, as "industry finds it difficult to suggest any alternative." I do not think the latter is true, as either industry or professional society would have no difficulty in drawing up a code which could be enacted into law and be acceptable to the great majority of manufacturers, engineers, contractors, and school boards. I admit there will always be a small minority you can never please.

We can all probably agree that the state laws, taken as a whole, have served the purpose for which they were originally drawn, and could probably be revised in a way to fall in line with certain current thought and knowledge. On the other hand, they still serve as a barrier preventing the use of untried or undemonstrated methods, and to that extent they still serve to safeguard the health of school children.

If confronted with the necessity of leaving the laws on the books as they stand or wiping them all out, I would infinitely prefer the former.

Correspondence

Does anybody truthfully believe that a body of students or individuals could be sealed in an air-tight room and continue to be healthy, comfortable, and efficient, even though means were provided to hold the temperature and humidity at a proper point? There are many old churches and old picture theatres, or even larger show-houses, which have been built years ago in which no provision whatever for any kind of ventilation was made. Did you ever step into one of these after it had been occupied for a while even though the temperature and humidity may have been kept within reasonable bounds?

Have I made the point clear that if only for common decency a certain amount of flushing the air contents of a room is imperative? I think so, from every aspect of comfort, health, and efficiency. Possibly many state laws still call for an excessive air change. If everyone agrees that they require some, the point at issue is how much. Doctor Winslow never advocated elimination of air supply—he merely introduced it in a different manner. So much for that.

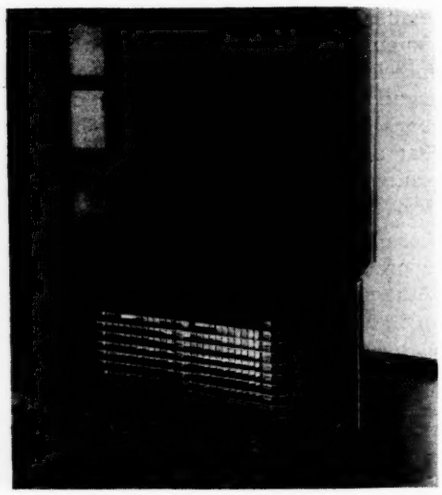
So far as equipment or mechanical means or devices are concerned, that should not be in the province of state legislation, rather should it confine itself to conditions to be effected.

Now laws can be of two kinds; one, in which conditions are definitely established and specified; the other, in which the determination of the conditions and the enforcing of them is delegated to some official of the state who has that activity in charge. The former is, of course, preferable providing there can be agreements as to the basis. The second method is admirable when the determination of the conditions is in the hands of adequately trained specialists of irreproachable honesty. It also possesses the virtue of being flexible to the extent that the rules can be changed to meet changing conditions, whereas the former requires new legislative action.

Inertia alone makes it difficult to change laws. Unless some one or some organization makes it their business to lobby or urge a revision in legislation, or new legislation, laws may remain on the books a long time after their usefulness has ceased to exist. No one individual, no one firm, or possibly no one industry, can afford the time and expense necessary to influence legislation along either more proper or more acceptable lines. It seems to me that the most appropriate organization to do this is one like the American Society of Heating and Ventilating Engineers, which represents the whole industry and includes among its members consulting engineers, manufacturers, operating engineers, contractors, and officials of school boards.

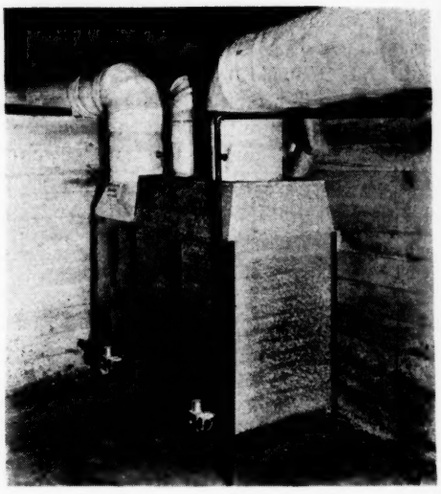
Several years ago, there was a committee appointed to draw up a code of minimum requirements on the heating and ventilating of buildings. In the beginning, it was hoped by some that the results of that committee's work could be used as a guide for legislative bodies in enactment of suitable legislation. Unfortunately, as they got deeper into it, the code took on more of the nature of good practices, rather than merely minima, and became utterly impracticable in form for that particular purpose.

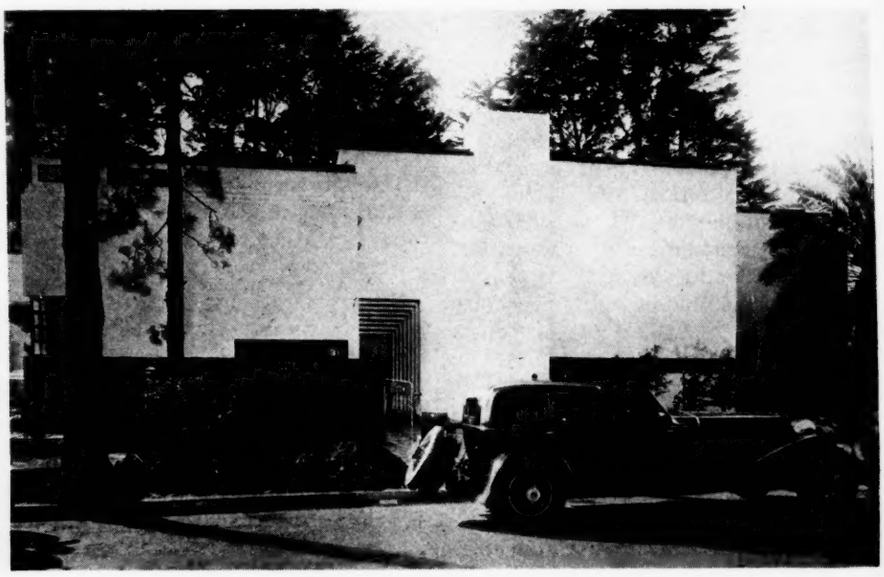
Notwithstanding the expressed opinion of certain engineers that we have been afflicted in recent years with



Home of Screen Star Heated with Gas

The modern residence of Dolores Del Rio, film star, and her husband, Cedric Gibbons, Art Director for Metro-Goldwyn-Mayer, and the designer of the house, is heated with a gas-fired warm-air furnace. A unique type of grille for warm-air supply is shown at the upper left.





Correspondence

"coditis," I firmly believe that a recommended form of state legislation to cover suitable requirements could be drawn up which could be subscribed to by the great majority of our society membership. Even that would not be enough to produce the results desired. It would be necessary to have a committee who could devote their time to lobbying this through their state legislatures as they convene, and this means spending money.

Short of the above full program if the money is not in sight, would be at least the first step, namely, of the drafting of the code, and this, I think an entirely appropriate action for our society to take.

W. A. Rowe.

Detroit, Mich.

Ventilation Laws

Editor Heating and Ventilating:

In regard to your editorial on Ventilation Laws in the September issue, I must say that I am not very strongly in sympathy with this for the reason that it breathes too much of the time-honored custom of criticizing schoolhouse ventilation laws, without pointing out the specific defects and suggesting the remedies which should be applied.

Your statement to the effect that much of the inertia in the matter of securing better and proper ventilation laws is due to the fact that the industry finds it difficult to suggest any alternatives is, in my judgment, entirely wrong. Such a statement as this leaves the question entirely open to another time-honored and apparently somewhat-cherished misinterpretation on the part of the average layman, schoolboard member, school officials, politicians and other pseudo-ventilation experts, to the effect that the whole question is so indefinite and befogged that one person's guess as to just what should be done is as good as another's.

This, as you well know, is entirely untrue, as the prescription for proper indoor atmospheric conditions has been more definitely determined and is more definitely written out today than are the prescriptions for most of the diseases for which we are treated.

This has been done by the Research Laboratory of the A.S.H.V.E. working in conjunction with the U. S. Bureau of Mines and the U. S. Public Health Service.

The best evidence that the proper methods of air conditioning have been developed is in the fact that these are now being used in practically all of the important commercial industries, in a large number of the commercial places of assemblage, in a great many public buildings and in our newer and up-to-date office buildings, both for the purpose of improvements in the processes of manufacture and for the health and comfort of the people within these buildings.

The classroom of a school building is no different from the working space of a bank or other commercial office and the assembly of a school is no different from the halls of Congress, as far as air conditioning problems are concerned.

For all of those who, if and when they are ready, really desire to know what to do about schoolhouse ventilation and for all of these State officials who wish properly to revise their state laws, there is a Code of Minimum Requirements for the Heating and Ventilating of Buildings, put out by the American Society of Heating and Ventilating Engineers.

This code does not specify the cubic feet of air to be handled or any particular method of handling it. As a matter of fact it does not even mention that old *fetish*—fresh air.

It sets up the proper standards to be maintained and suggests many general methods of securing them. It allows recirculation and air resuscitation, humidification, cleaning, ozonizing, ionization and whatever may be necessary in actually maintaining proper air conditions.

If, in addition to this, the State and other school officials would employ competent engineering services properly to interpret and put such codes into effect, the whole problem would quickly be solved.

Now as to what is wrong with existing State ventilating laws, they are not so basically wrong as some would have us believe. The circulation of 30 cu. ft. of air per min. per pupil is substantially what is required in the average classroom for proper air motion and distribution. Just bringing this quantity of air in from the outside and heating it does not produce proper air conditions. It must be clean, be at the proper temperature, have the proper degree of humidity, be properly distributed and be delivered so as to produce agreeable air motion without drafts. These are the ingredients which constitute modern air conditioning, but are not included in existing laws because they were not properly understood at the time these laws were drawn.

The only important question is, as to whether this should all be air from out-of-doors or whether it should be partially recirculated and reconditioned, and to what extent. The answer is that it may be either but, in the interest of economy, recirculation with air filtration, ionization and humidity control may be used, ranging from no recirculation in mild weather (when the air does not have to be heated) up to 75% recirculation in zero weather. All under automatic temperature, humidity and recirculation control and with air filters, air washers and ionization.

In assembly rooms the same applies, except that for crowded assemblages for use in mild or hot weather, air cooling and dehumidification should be used.

Such a system can be installed for 75% of the cost and can be operated for less than 50% of the cost of systems now generally required by State laws.

One more important point is that most existing State laws require systems to be installed, but do not require them to be operated. Any new laws should require that systems be operated during all sessions and in such manner as to produce the minimum air conditions prescribed and State inspection should be provided to see that this is done.

Now as to whether the heating and ventilating industry will lead or follow—it is already leading and the question is as to when those who really want proper indoor air conditions will follow.

Perry West

Newark, N. J.

Addison H. Beale

Addison H. Beale, president of A. M. Byers Co., Pittsburgh, Pa., died suddenly on October 28 at the home of one of his sons in Chicago, while on a business trip to that city.

Mr. Beale first became connected with the A. M. Byers Company in 1925 as president, starting on a large modernization program which culminated with the completion of the Byers new plant on October 8, 1930.

Mr. Beale was in his 62nd year at the time of his death, and his wife, two sons and a daughter survive him.

Richard T. Keefe

Richard T. Keefe, well known to the heating industry in the Boston area, died on November 6. He was for many years identified with the E. T. Keefe Company, a member of the Heating and Piping Contractors Boston Association serving for a time as its president and treasurer. During recent months Mr. Keefe had severed his connection with this company and had joined the Badger Company.

Mr. Keefe, who was forty years of age, is survived by his wife and four children.

In the article "Duct Design for Fan Steam Systems" in the December issue, the head H given on page 86, fourth line, should have been in feet of air, and not feet of water.

State Laws Relating to Heating and Ventilating Industrial Buildings

(*Continued from December Issue*)

Laws Relating to Specific Industries and Processes.

Nine states have, in addition to their general ventilation laws applying to industrial buildings, laws relating to various specific industries, or processes, such as foundries, grinding, etc. Abstracts of such laws follow.

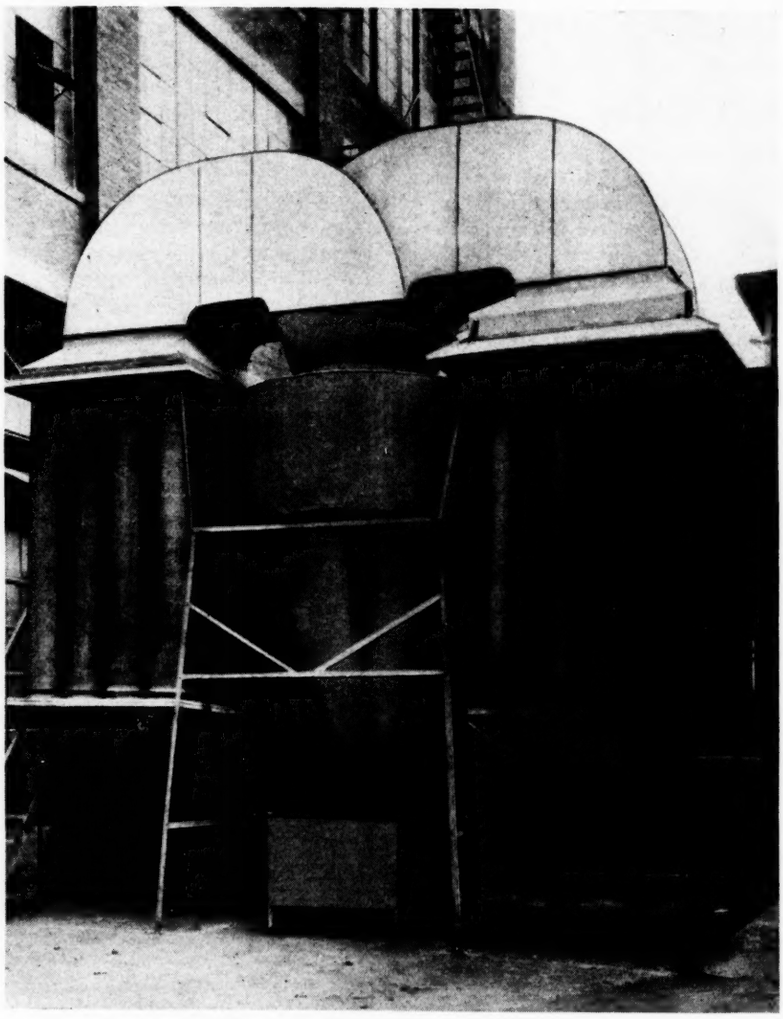
Bakeries

Sec. 6843, Sanitation of Bakeries, etc., of the Missouri laws states in part:

Every building or room occupied as a biscuit, bread or cake bakery or confectionery shop shall be constructed with air shafts, windows or ventilating pipes sufficient to insure ventilation. Every room or rooms used as a bake shop or confectionery shop wherein food stuffs are mixed, manufactured or baked, and all troughs, mixing boxes, steam boxes and other appliances used in the preparation of food stuffs shall be left open, ventilated and aired for a period of not less than twelve consecutive hours during each week between the hours of 6 p.m. on Saturday and 6 a.m. on Sunday, and during these twelve hours no sponge setting, mixing or baking shall be done in the shop.

Act. No. 325, July 9, 1919, laws administered by the Department of Labor and Industry of Pennsylvania provide essentially for the following:

All bakeries shall be so constructed that at least 200 cu. ft. of fresh air per min. will be admitted to that part of the bakery where one and not more than three persons are employed in the manufacture of such products. If more than three persons are so employed therein, then for each such person over three, spending three hours or more at a time therein, the quantity of air admitted shall be increased by at least 66 cu. ft.



Dust Bag Separator for Reclaiming Fine Dust. This Installation is One in a Rubber Plant

Industrial Code Bulletin No. 4 of New York, states, in effect:

Wherever bakery or confectionery products are fried in fat, or candy is boiled over an open stove, a ventilating hood and pipe shall be provided which shall effectively take off the smoke, gases and vapors; the pipe shall not be less than 4 in. in diameter and shall be attached at the extreme top of hood; the hood shall be cone shaped, shall not be raised more than 6½ ft. from the floor and its width and breadth shall at least equal the width and breadth of the stove to be ventilated.

Oven ashpits shall be provided either with flue and damper or with ventilating hood and pipe not less than 4 in. in diameter leading either to the outer air and extending to a point at least 20 ft. higher than the top of hood, or leading to the flue of the building.

Oven doors shall be provided with a ventilating hood of at least the same width as the doors and with pipe not less than 4 in. in diameter leading either to the flue of the building or to the outer air and extending to a point at least 20 ft. higher than the top of the hood, except when indirect heating ovens are

used, when the steam flue of the oven is 8 in. or more in depth, or when there are over the oven either louvered openings or pivot swings in skylights of 4 sq. ft. area per oven or satisfactory mechanical means of ventilation.

Whenever a hood is used in complying with the requirements of the second and third paragraphs, it shall be constructed of brick, cement, or metal, shall extend not more than 1 ft. from the oven wall and shall be cone shaped with pipe attached at extreme top of hood.

Brewing and Bottling

Regulations for Brewing and Bottling, 1930 edition, of Pennsylvania call for:

Excessive amounts of foul air or steam vapor in workrooms shall be removed by an approved system of ventilation.

Celluloid

The regulations of New Jersey covering the manufacture of celluloid provide for sizes of pipe dependent upon the sizes of wheels; also cover mixing mills, and give regulations for tests, proportions, and gauges of metal.

Cereal Mills, Etc.

Regulations for Cereal Mills, Malt Houses and Grain Elevators of Pennsylvania, provide essentially for the following:

All belt conveyors for receiving grain shall be provided at the point of charge, with a hood connected with an exhaust fan of sufficient capacity to remove or prevent the escape of all dust arising from the conveyor.

Dust rooms or stive rooms, shall not be used. A fan discharging dust shall discharge either to the outer air at such points where the dust cannot reenter the mill, or an adjacent building, or to a dust collector, and the air vent or outlet to every cyclone dust collector, shall discharge to the outer air or to another dust collector. All tubular dust collectors shall be maintained in a proper state of repair so as to prevent leakage of dust into the room. A fan which discharges into any dust collector shall be kept in operation at all times, while the machine or machines with which it is connected are in operation. There shall be no direct connection between dust collectors and boiler rooms or incinerators. This rule also applies to malt house elevators.

Mills and all buildings, structures, and elevators, used in connection therewith shall be kept thoroughly clean and free from dust.

The piping of all exhaust systems and all ducts shall be kept tight.

Dry Color

Regulations of Pennsylvania for the dry color industry provide substantially the following:

All crushing mills, grinding mills and sieving machines operating on material in a dry state, if containing lead in excess of 10%, shall be connected with an efficient air exhaust and dust collecting systems.

All mixing and shading of all dry materials containing lead in excess of 10% of arsenic, shall be performed in a room or rooms with mechanical exhaust ventilation.

Dyeing and Cleaning

The Industrial Code, Bulletin No. 29, of New York State, relating to dry dyeing plants and dry cleaning plants, substantially calls for the following:

Each drying tumbler shall be ventilated to the outer air by means of properly constructed pipes and ducts connected to an exhaust fan or fans to remove dust, gases, vapors, fumes or fibres generated by process. Such fan shall lead directly to a discharge pipe carried to a height of at least 6 ft. above the roof. No discharge pipe shall terminate within 10 ft., measured horizontally, of any door or window in a factory building or any building wherein a mercantile establishment is conducted. The fan shall be properly housed and kept running constantly while drying tumbler is in use. The fan blades and spider shall be constructed of non-ferrous metal, the air supply shall be taken from outside of building.

Under "Ventilation of Wet Dye Houses or Rooms" the regulations are as follows:

"Excessive heat" shall be construed to mean a dry-bulb temperature in a wet dye house or room in excess of 80° F. when the wet-bulb depression or difference in reading between the dry-bulb and wet-bulb temperatures is less than 8°, and the outdoor temperature is 70° F. dry bulb or less. It shall also mean, when the workroom dry-bulb temperature exceeds by 10° or more the dry-bulb temperature recorded outdoors when the outdoor temperature is more than 70° F.

"Excessive humidity" shall be construed to mean a moisture content of air within the wet dye house or room when the wet-bulb depression or difference in reading between the dry-bulb and wet-bulb temperatures is less than 8° F. and the outdoor dry-bulb temperature is 70° F. or less.

Every wet dye house or room in which people are regularly employed in a dry cleaning plant and in which any machinery or apparatus is operated generating steam, vapors or radiating excessive heat shall be ventilated at all times during working hours by an approved method of natural or mechanical ventilation or both, as may be necessary to maintain proper and sufficient ventilation, proper temperature and humidity and to reduce excessive heat.

Felt Hatting Industry (Back and Front Shop)

Regulations for this industry in New Jersey call for the following:

All blowing machines should be fed by means of an automatic device, and all dust and fur fibre liberated during the blowing and cleaning operations should be removed from the air of the workroom either by means of the exhaust system standardized by the Bureau of Hygiene and Sanitation or other approved and equally efficient device.

The temperature of the blowing room (when due to artificial heat) should not be allowed to exceed 75° F., and when live steam is employed to increase the humidity, excess temperature should be prevented by means of a cold water spray, or other efficient device.

The temperature of the mill room (when due to artificial heat) should not be allowed to exceed 75° F; and when live steam is employed to increase the humidity, excess temperature should be prevented by means of a cold water spray or other efficient device.

All steam generated in the operations of starting, first and second sizing, dyeing and blocking, should be controlled and eliminated by means of efficient mechanical exhaust ventilation.

Wood alcohol fumes should not be permitted to pollute the air of the workrooms either during the shellacking or drying operations. Efficient means for their removal should be installed and care exercised to safeguard against the hazard of explosion.

All dust generated in the operation of pouncing and sand papering machines should be controlled and eliminated by means of the mechanical exhaust ventilation system standardized by the Bureau of Hygiene and Sanitation.

All steam pots and singeing flames should be hooded and have direct connection to the outer air by means of pipes not less than 12 in. in diameter, with efficient natural draft ventilation.

Wherever gas is employed for ironing purposes, the health of the worker should be safeguarded by means of efficient mechanical ventilation.

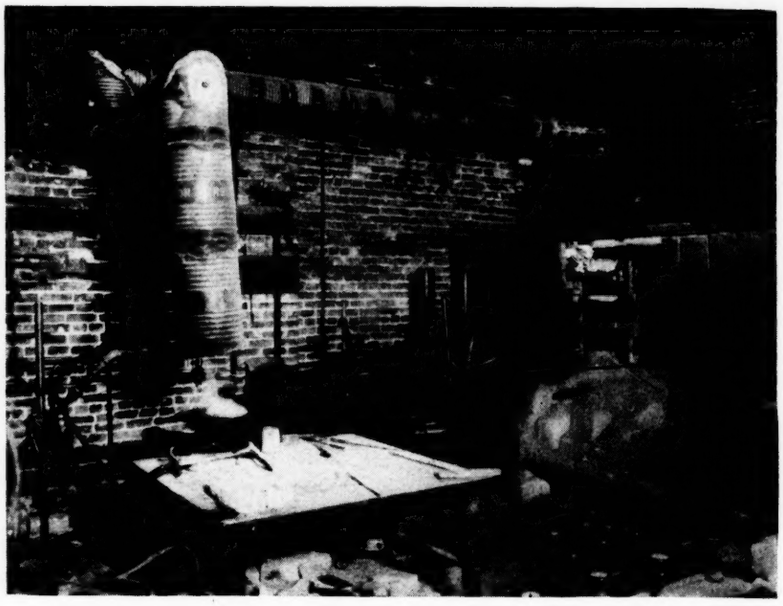
When the grade of hats handled makes powdering necessary, the dust generated should be drawn away from

the workers by means of efficient exhaust ventilation.

To control the excessive heat generated in the flanging and matricing operations, ample ventilation should be provided by means of windows, wall fans, or other devices; all steam plates should be covered on bottom and sides with asbestos to prevent direct radiation and wherever possible an electrical device for heating the flange-bags should be installed.

Sizes of pipe are given for crown pouncing and frame pouncing processes and standard pipe sizes for removing steam from sizing machines, starting tanks, block, brim, tip and dye kettles are given for various diameters of kettles.

Tests and proportions are included in the regulations.



Carbide Drums Welded Together to Form Exhaust Ventilating Ducts

Felt Hatting Industry (Fur Preparing)

To make fur cutting a reasonably safe and sanitary occupation, New Jersey has provided that shops shall observe the following requirements: Enclosure, and standard exhaust ventilation on tumbling drums; standard exhaust ventilation for the fur-plucking machine; mixing the carrot outside the shop; storing the carrot in ventilated chambers; rubber gloves and impervious aprons for the carrotter; a drained bench of impervious material slanting away from the carrotter; ventilating hood with efficient exhaust above the carroting table for both hand and machine work; efficient draft to carry off all fumes from drying ovens; standard exhaust ventilation on the brushing machines; exhaust ventilation on the cutting machines; mechanical exhaust ventilation on all blowing machines; removal of steam by means of ventilating booths from all boiling tanks in the treatment of "rounding"; mechanical exhaust ventilation on all tearing machines, teasing blowers, choppers and grinders; no raising of the temperature of the workroom above 60° F. by means of artificial heat; abundant natural or mechanical ventilation in the storeroom for sacked fur, and wherever "white-carrotted" skins are drying.

Sizes of pipes for the following machines are given: Pulling machines, shearing machines, brushing machines, cutting machines, tumbling cages, and fur blowing machines. Proportions and test regulations are included.

Flint Grinding

The Standard and Engineering Industrial Standards of New Jersey contain the following: Dust generated in the process of grinding flint rock, feld spar, and limestone must be removed by an exhaust system.

The standards cover both crushers and ball grinders, and the regulations cover tests, proportions and gauges of metal.

Foundries

Industrial Bulletin Number 10, Rules and Regulation Relating to Safe and Sanitary Working Conditions in Foundries and the Employment of Women in Core Rooms, published by the Department of Labor and Industries, Division of Industrial Safety for Massachusetts, states:

Sec. 4. Where smoke, steam, gases or dust arising from any of the operations of the foundry are dangerous to health or eyes, and where a natural circulation of air does not carry off such smoke, steam, gases or dust within reasonable time, then the impurities shall be removed by mechanical means.

Sec. 5. Castings shall be cleaned or shipped in separate cleaning rooms or in sections of the shop exclusively devoted to the cleaning and chipping of castings. It is, however, understood that large castings may be cleaned in the foundry proper, provided adequate means are furnished for the removal of dust, smoke, steam or gas.

Sec. 6. Exhaust systems shall be installed to carry off effectively the dust arising from the operation of tumbler mills, except where the mill is operated in an adequate enclosure or outside the foundry.

Sec. 10. Proper and sufficient heat shall be provided and maintained in every foundry. Heaters must be connected with the outer air. Open fires may be used for the drying of molds or cores if coke containing less than 1½% of sulphur is used; charcoal, gas or oils also may be so used; such drying of molds or cores shall be done at night where practicable.

Regulations relating only to brass foundries state essentially that:

Sec. 31. The regulations relative to dust, smoke, gases or fumes, ventilation, heat, cleaning rooms, drying and locker accommodations, as specified for iron foundries, shall apply to brass foundries.

Sec. 35. Hoods shall be provided directly above all brass-melting furnaces using gas or oil as fuel, which will effectually trap all gases and fumes generated in the melting of the metal; such hoods shall be provided with flues or ducts to lead the gases or fumes to outer air. Ventilators shall be provided over all other furnaces used for melting brass or composition metal to effectually remove the gases above the furnaces.

Sec. 5347-8341 of the laws of Michigan states essentially:

Whenever a foundry is so constructed or operated that smoke, steam, dust or noxious gases are not promptly carried off by the general ventilation, exhaust fans shall be provided. No salamanders or open fireplaces shall be used, unless ample provision be made for conveying the gases arising therefrom directly from the building. Foundries shall be reasonably well heated during the cold and inclement weather.

Sec. 6814 of the Labor, Industrial and Statistical Laws of Missouri, 1927, states in effect:

Every corporation, company, or person engaged in operating a foundry shall provide and maintain adequate and effi-

cient devices for carrying off all poisons or injurious fumes, gases and dust from the foundry.

Bulletin No. 10 of the Industrial Code of New York, covering foundries and core rooms, includes the following:

Smoke, steam or gases generated in foundries shall be effectively removed. The milling and cleaning of castings and the milling of cupola cinders shall be done under such conditions to be prescribed by the board as will adequately protect the employes from dust. Use of heaters discharging smoke or gas into the workrooms is prohibited except that open fires may be used under conditions prescribed by the board in its rules.

Rule 556. Where smoke, steam, gases or dust arising from any of the operations of the foundry are dangerous to health and where a natural circulation of air does not carry off such impurities, there shall be installed and operated mechanical means of ventilation approved by the Commissioner of Labor.

Rule 559. Where tumbler mills are used, exhaust systems shall be installed to effectively carry off the dust arising from the cleaning of castings, except where the mill is operated outside the foundry.

Laws of New York relating to brass foundries state in effect that:

Rule 587. The rules relative to dust, smoke, gases or fumes, ventilation, sanitation, heat . . . . as specified for iron and steel foundries, shall apply to brass foundries, etc.

Rule 592. Hoods shall be provided directly above all brass melting furnaces using gas or oil as fuel, which will effectually trap all gases and fumes generated in the melting of the metal; such hoods shall be provided with outlet pipes to lead the gases or fumes to outer air.

Ventilators shall be provided over all other furnaces used for melting brass or composition metal, to effectually remove the gases above the furnaces.

Laws of Pennsylvania administered by the Department of Labor and Industry, state that:

The said toilet room and the said water closet shall be connected with the foundry building in such a way that access thereto may be had without exposure to the open air and shall be properly heated, ventilated, cleaned and protected, so far as reasonably practicable, from the dust of such foundry.

Labor Laws of the Commonwealth of Virginia administered by the Department of Labor and Industry, 1930, contain Sec. 1833, which provides essentially for the following:

Every person, firm or corporation employing men to work in any foundry or molding shop shall provide in each workroom proper means of ventilation. If excessive heat, or if steam, gases, vapors, dust or other impurities that may be injurious to health be generated, the room must be ventilated in such manner as to disperse the steam, gases, vapors, dust or other impurities, and render them harmless.

Fur Dressing

The laws of New Jersey provide for sizes of pipe for dust exhausts in fur dressing establishments, which are given for tumbling cages, hair plucking machines, and shearing machines. Copper oxides, sulphate of iron, and amido phenol must be weighed out under exhaust hoods. Steam cooking tubs must be hooded and provided with exhausts. Regulations covering tests and proportions are given.

Grinding, Buffing and Polishing

Labor Laws of the Commonwealth of Virginia, administered by the Department of Labor and Industry, 1930, contain Sec. 1821-a, which states essentially that:

Grinding, polishing and buffing wheels must be equipped with suitable devices for the protection of operators and other persons from dust and refuse thrown off from such wheels. All persons, firms, and corporations operating a factory, or other place or building where such wheels are used in the course of the manufacture or the working on of articles of the baser metals, shall provide the wheels with a hood and exhaust system.

Every hood shall be made of metal or other suitable material and be of such form and so located in relation to the grinding surface of the wheel that the dust and refuse will fall or be drawn into the hood and be carried off by the pipe attached to it, and so as to prevent injury to the operator if the wheel shall burst. Connection of such hood with an exhaust fan or other suction device shall not be required when less than five of such wheels are owned or operated by one person or concern; when such wheel is provided for only occasional use by workmen in grinding the tools used by them, or when water is used upon such wheel at the point of grinding contact.

Every hood shall be so constructed as to expose the smallest portion of the wheel consistent with efficient operation, and its free edges shall be turned back or faced to prevent injury to the hands of workmen. Every such fan or other suction device shall be kept in constant operation while the wheels are in operation.

The Sanitary and Engineering Industrial Standards of New Jersey contain tables of sizes for the branch pipes, depending on the size of the rag or hard wheel, and cover complete detailed specifications for the construction of hoods for buffing and polishing wheels, including data on materials, gauges, proportions, etc.

Minnesota's Minimum Requirements Governing the Installation of Exhaust Systems for Buffing or Emery Wheels are substantially as follows:

Before installing any exhaust system, a sketch should be submitted to the Industrial Commission, showing layout and size of piping. Power and capacity of blower or exhaust fans should be given. Pipes should not be longer than forty times their diameter. If bends are used, elbows should be used that are made on a radius of not less than 1½ times the diameter of the elbows.

The area of any main duct into which a number of branch pipes enter, shall not be less than the combined areas of the branches entering into it, plus 20%. Wherever possible, the main duct should be below the emery or buffing wheels. The area of the opening on the suction side of any blower shall not be less than the area of the main duct leading into it. No branch pipe shall enter the main duct at an angle greater than 45°; 30° is preferred.

Vertical runs are undesirable. If unavoidable, a trap should be installed at the bottom of the vertical pipe, then all material which is not elevated by the suction of the fan or blower, will fall into the trap from which it can be removed at regular intervals.

The diameter of pipe leading from the exhaust hoods of buffing or grinding wheels shall not be less than 4 in.

There should be an air velocity of not less than 3000 ft. per min. at the hood opening of buffing wheel or grinder.

The exhaust system should be of sufficient power to remove all dust and waste from buffing or grinding wheels.

Laundries

In the requirements for laundries in New Jersey the following are included:

Where steel plate exhausters are used, the main duct at all cross-sectional points shall have an area at least 20% greater than the combined areas of branch pipes entering the same.

Where disc or propeller fans are used, the size of the main duct shall be increased in accordance with conditions surrounding each installation; and the

discharge of the fan so protected that conflicting air currents will not interfere with the efficiency of the system.

Clean-out hand holes shall be placed on 8-ft. centers.

Flat work ironers (cylinder type) shall be hooded, and provided with two exhaust openings, equally spaced. Each opening to have an area of at least 200 sq. in.

Chest type 100 in. single roll ironer shall be hooded and provided with two exhaust openings, equally spaced. Each opening to have an area of 78 sq. in.

Flatwork ironers (chest type) shall be hooded and provided with two exhaust openings, equally spaced, having the following areas: Machines having 4 to 6 rolls shall have two exhaust openings of not less than 150 sq. in. each.

Machines having 8 rolls shall have two exhaust openings of not less than 200 sq. in. each.

Collar ironers: Where the roll is 40 in. long, the exhaust opening in the hood shall have an area of 80 sq. in.

Cuff and Neck Band Presses: Each press shall be hooded and provided with an exhaust opening having an area equal to 28 sq. in. for each individual press.

Sizes for exhaust openings for various size rolls for rotary ironers are given.

A velocity of air of at least 1000 lin. ft. per min. shall be generated and maintained at the point where the exhaust duct connects with the hood. An opening properly capped shall be provided at this point to permit a test being made with an anemometer. Test shall be made with the entire system open and unobstructed.

Industrial Code Bulletin No. 27, of New York, relating to the Construction, Equipment, Maintenance and Operation of Laundries, essentially provides for:

Every workroom in which people are regularly employed in a laundry and in which any machinery or apparatus is operated generating steam, vapors or radiating excessive heat, shall be ventilated by an approved method of natural or mechanical ventilation or both, as may be necessary to maintain proper and sufficient ventilation, proper temperature and humidity, and to reduce excessive heat at all times during working hours.

This rule is followed with complete specifications for the systems to be used in connection with the specific machines.

Pennsylvania's regulations for laundries call for exhaust or ventilating fans as follows:

Each exhaust or ventilating fan except fans in drying rooms, any part of which is within 7 ft. of the floor or a working platform, shall be completely covered with wire mesh of not less than No. 16 gauge with openings not larger than ½ in. at their greatest dimension.

Leather Work

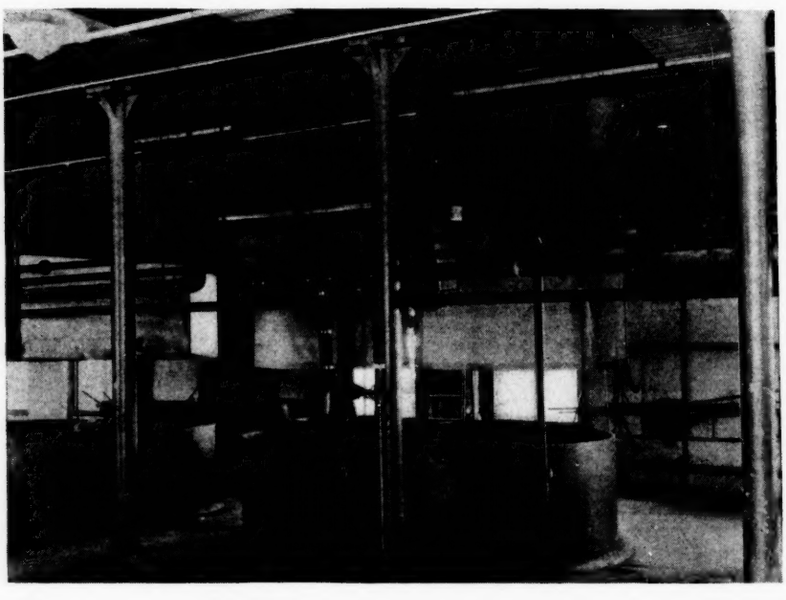
Standard regulations for removing dust generated in the manufacture of leather are provided by New Jersey.

Sizes of pipes for exhaust systems connected with shaving machines, buffing machines of various sizes are given, together with regulations covering tests, proportions, and gauges of metal.

Metal Dipping

New Jersey's standard regulations covering fumes generated in industries where metals are dipped in acid or alkali solutions, state that a main duct shall have a cross-sectional area at least equal to one-half the combined area of exhaust openings; bottom of main duct shall be 7 in. wide. Front shall incline at an angle of 115° to the bottom; exhaust openings with an area equal to 25% of the area of the liquid giving off the fumes shall be located as near the point of origin of the fumes as the nature of the work will permit; a velocity of air of 1,000 lin. ft. per min. shall be generated in each exhaust opening; if sheet metal is used in the construction of the duct, an acid resisting paint should be used to prevent corrosion; exhaust fan should be treated with acid resisting paint to prevent corrosion.

Plans will be furnished by the Bureau of Hygiene and Sanitation for the following purposes: Color blowing in potteries; glaze blowing in electric porcelain plants; varnish blowing; lacquer blowing; shellac blowing; paint blowing; linotype machines; stereotype machines; sizing machines (hat factories); pouncing machines (hat factories); flat work ironer (chest type); flat work ironer (cylinder type); rotary gas ironer; single roll ironer; collar roll ironer; cuff press, neck band press, lead casting; acid and alkali fumes; buffing wheels; dust hood for metal grinding; dust hood for pearl grinding; showers and lavatory washing layout; toilet room layout; toilet room vestibule; and natural draft hood.



Installation of Exhausters in Chemical Plant

Paint Grinding

Pennsylvania includes in its laws regulations for paint grinding substantially as follows:

All packages 100 lb. in weight or less may be packed by hand under a hood equipped with an efficient air exhaust, which shall be connected with an efficient dust collecting system, such system to be regulated by the discharge of air from a fan, pump, or other apparatus through an enclosed cloth dust collector, having an area of not less than 1 sq. ft. of cloth to every cubic foot of air passing through it per minute. If this cloth dust collector is not of the portable type, it shall be placed in a separate room, or in a permanent dust house equipped with baffles, or such other apparatus as will adequately take care of all dust which it may receive.

(To be continued)

Activities of Heating Board of Trade Featured at Building Congress

THE organization, purposes, and accomplishments of the Heating Board of Trade of New York were explained to the members of the New York Building Congress by Pres. Werner Nygren and Col. D. H. Sawyer at a luncheon held Thursday, January 22, at the Hotel Commodore, New York.

In his talk, Mr. Nygren told the audience that the members of the Heating Board of Trade realized that they had a difficult job laid out for them. After one year's work, it has hardly scratched the surface. However, in his opinion, the contact among the members is building up an understanding which will result in various competitive groups working together more harmoniously in the future. The board has worked up methods by means of which it is expected that "shopping of bids" will be curbed and a plan whereby the names of those owing money to members of the board will be published to all the members. This is expected to reduce appreciably the losses due to manufacturers and jobbers carrying irresponsible contractors on their books, and to reduce to some extent at least, destructive competition.

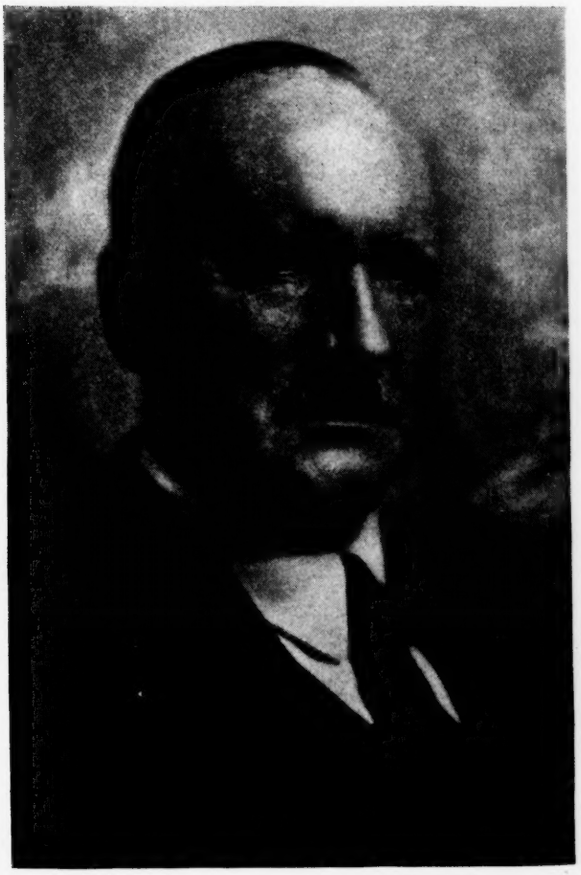
Although plumbers and representatives of other mechanical trades are not included in the membership of the board, Mr. Nygren said that some day it may be necessary to include these trades. The board is not attempting to build up a monopoly or to reduce legitimate competition, but merely to build up means by which the contractor can make a fair profit.

Mr. Nygren was followed by Col. D. H. Sawyer, executive secretary of the board, who outlined the business situation at present and compared it with conditions thirty years ago, at which time, due to the organization of powerful trusts, business became so bad that it was necessary to pass legislation to curb the activities of the big business firms.

According to data worked up by the Heating Board of Trade, the construction industry in New York totaled $57,000,000 in 1930, as compared with $74,000,000 in 1929, a reduction of 23%.

Among the activities which the Heating Board of Trade will investigate are the establishment of quantity surveys, shopping of bids, the publishing of names of those owing overdue accounts, an attempt to confine the production of plans and specifications to legitimate architects and engineers, and others.

During the luncheon, the board was referred to as an infant, although a husky one, and the luncheon, which was well attended, marked the first public appearance of the board.



Werner Nygren

Colonel D. H. Sawyer

News of the Month

Heat Flow Division of A. S. M. E. Is Planned

At the conference held by the A.S.M.E. on December 5 on the question of organizing an Air Conditioning Division was brought out the almost unanimous opinion that what was really needed was a Division on Heat Flow or Heat Transmission.

W. H. Carrier, chairman of the conference, as well as a number of others, strongly objected to a Division on Air Conditioning, since they doubted that it would be possible to avoid duplication with the American Society of Refrigerating Engineers and the American Society of Heating and Ventilating Engineers. There was, however, a unanimous opinion on the need of a division in the society to deal with the more technical aspects of heat transfer.

It was pointed out that a Heat Flow Division would not only be of great advantage to the society but might also be a joint activity with, or at least have the joint cooperation of, the American Society of Refrigerating Engineers, the American Society of Heating and Ventilating Engineers, and the American Institute of Chemical Engineers. There was some comment on the fact that the Applied Mechanics Division considered within its scope the highly technical aspect of thermodynamics, since many thought this subject might better be discussed in the proposed Division on Heat Flow or Heat Transmission.

H. W. Brooks, Chairman of the Standing Committee on Professional Divisions, stated that the society had no intention or desire to duplicate the work already being done, but that the A.S.M.E. would be glad to consider the formation of a possible Heat Division if cooperation with other interested organizations were assured.

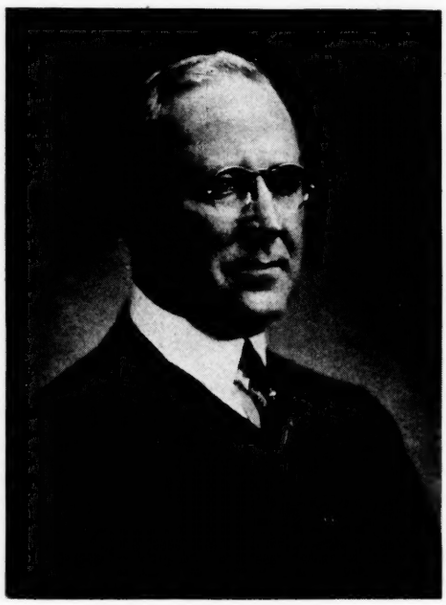
Mr. Brooks then made a motion asking that each of the three societies previously mentioned appoint two representatives to consult with two representatives of the American Society of Mechanical Engineers as to the advisability of the formation of a Division on Heat Flow, and determine to what extent their societies might possibly cooperate in its activities. The motion was unanimously passed by the conference.

Those invited to attend the conference had either urged the society to foster activities dealing with such topics as air machines, heating and ventilating, refrigeration, air conditioning, and thermal engineering, or else were A.S.M.E. members in and around New York who were also members of the above mentioned societies.

Hudson County, N. J., Adopts Smoke Ordinance

On January 20 the Hudson County Board of Health and Vital Statistics adopted a smoke ordinance, prohibiting the making of dense smoke within the limits of Hudson County, New Jersey, and establishing a Department of Smoke Regulation. The ordinance is based on the standard smoke ordinance of the U. S. Bureau of Mines, which was compiled by Dr. O. P. Hood, chief engineer of the bureau, and an authority on smoke abatement.

The new ordinance was drawn up under the direction of Col. Elliott H. Whitlock, former Smoke Commissioner of Cleveland, Ohio, and now in charge of smoke abatement research at Stevens Institute of Technology, Hoboken, New Jersey, and William G. Christy, former executive secretary of the St. Louis Smoke Abatement League and recently appointed Smoke Abatement Engineer of Hudson County.



Col. Elliott H. Whitlock
Director of Smoke Abatement Research, Stevens Institute of Technology, Hoboken, N. J., who Assisted in Planning the New Smoke Ordinance Recently Adopted by Hudson County, N. J.

Col. Whitlock made a remarkable record in eliminating smoke in Cleveland, and is considered an expert on smoke abatement work. He commended the officials of Hudson County for tackling the smoke problem and stated that this is the first time in the history of the United States that a county government has passed a smoke ordinance and set up a smoke department.

The ordinance creates a Department of Smoke Regulation under the Board of Health and Vital Statistics in charge of a smoke abatement engineer. In drawing up the ordinance, Colonel Whitlock and Mr. Christy had the assistance of H. K. Kugel, acting smoke commissioner of Cleveland. Mr. Kugel is Chairman of the Smoke Abatement Committee of the Fuels Division of the A.S.M.E. He recently collected and compiled smoke ordinance from every American city having such an ordinance. This collection comprises over 200 smoke ordinances.

Dense smoke is defined in the ordinance as being equivalent to No. 3, or greater on the Ringelmann Chart, or as being so thick at the point of emission that it cannot be seen through. The ordinance prohibits the production or emission of dense smoke, fly-ash or fumes from any smoke stack or open fire, except that from a locomotive or steam boat, for a period or periods aggregating two minutes in any period of fifteen minutes. It also prohibits smoke equivalent to No. 2 of the Ringelmann chart from any stationary plant or open fire for a period or periods aggregating twelve minutes in any period of one hour. No. 2 smoke is defined as being so dense that it can be dimly seen through.

BOILER OR FURNACE REPAIR WORK REQUIRES PERMIT UNDER NEW ORDINANCE

The ordinance also requires that any person or corporation constructing, or reconstructing, altering or repairing any furnace, boiler, stack, or any apparatus connected with a stack, shall secure a permit for such work from the smoke department. An advisory board of three engineers is created. This board acts as a board of appeals, and anyone applying for a permit may appeal from the decision of the smoke abatement engineer to the advisory board.

Any person or corporation violating any prohibition or requirement of the new smoke ordinance is subject to a fine not exceeding fifty dollars or imprisonment in the county jail or workhouse for not more than thirty days, or both. Each day's violation constitutes a separate offense.

News of the Month

Chicago Engineers Hear Talk on Aerology

Dr. E. Vernon Hill spoke on the subject "Aerology and Eternity—the Physiological Possibility of Air Conditioning" before the regular monthly meeting of the Illinois Chapter of the A.S.H.V.E., held January 12 at the Hotel Sherman, Chicago.

Following a discussion of Dr. Hill's topic, the chair introduced a new feature to the chapter, namely, a question box; the idea being that any member may send in to the secretary a question which he would like to have discussed at a meeting. The first question presented was: "What is the proper pipe to use for steam returns under a floor?"

Nygren to be 1931 President of Heating Board of Trade of New York

At the annual meeting of the Heating Board of Trade of New York, January 14, the following officers were elected: president, Werner Nygren; vice-presidents, R. A. Wolff, A. P. Keasbey and S. C. Carrier; treasurer, P. R. Mork.

The report of the secretary showed that the membership had increased from 80 to 122, with very few resignations due to the business depression.

The Finance Committee recommended, and the Board of Governors approved, that the budget for 1931 should not exceed $20,000. The Board of Governors in considering the credit situation is working on the problem, and it is expected that an arrangement will be worked out whereby all members will submit a list of overdue accounts, the list to be available to all members in order to improve the credit situation. The time limit after which an account will be listed as overdue will depend on conditions in the particular branch of the industry the seller represents.

The Board of Governors for 1931 will be composed of the following:

Engineers: L. T. M. Ralston, A. L. Jaros, Jr., J. G. Eadie.

Contractors: L. K. Berman, W. W. Hubbard, J. L. Murphy.

Sub-Contractors: K. Duff, George Allen, Jr., B. W. Bennett.

Manufacturers: G. E. Olsen, H. B. Hedges, E. J. Ritchie, William LeCompte, J. J. Ney.

Jobbers: B. F. Gilmour, M. M. Goldsmith, E. F. Keating.

Utilities: F. T. Head.

Hildebrand Elected Secretary-Manager of Plumbing and Heating Industries Bureau

Joseph G. Hildebrand, director of regional management conferences for the Plumbing and Heating Industries Bureau, was elected secretary-manager of the bureau at a meeting of the executive committee at the Congress Hotel, Chicago, January 6. Mr. Hildebrand succeeds E. L. Flentje.

Joseph G. Hildebrand

In the selection of Mr. Hildebrand, the executive committee recognized the executive ability, the energy and the enterprise which he demonstrated in the direction of the fourteen regional management conferences which he has conducted for the bureau in various cities in the Middle West since July 1. The total attendance of contractors, employees, wives of contractors, and representatives of wholesalers and manufacturers at these conferences was nearly four thousand.

Mr. Hildebrand had a varied experience in the plumbing and radiator heating industries. He began by working with the tools as a plumber in a New England town. Later he had a plumbing contractor business of his own. Subsequently, he became manager of a wholesale branch. He held this position for eight years, after which he became sales promotion manager for a wholesale-manufacturing firm in the East.

Plumbing and Heating Salesmen Meet in New York

The National Association of Plumbing and Heating Salesmen held its annual convention December 19, at the Hotel Governor Clinton, New York.

Clarence Popp, New York manager of the Toledo Pipe Threading Machine Company, was the speaker of the evening at the November meeting of the New York Chapter, held at the Hotel Governor Clinton on November 19. He delivered an interesting talk on modern salesmanship. John H. Taylor then addressed the meeting on the past, present and future of industry.

Field Engineer Engaged by A.O.B.A.

Charles A. Whitney became a member of the technical staff of the American Oil Burner Association on January 1, according to an announcement by Harry F. Tapp, executive secretary. Mr. Whitney will spend most of his time in the field working with fire chiefs, and municipal authorities in the drafting and modernization of oil burner ordinances.

A graduate of the University of Kansas, Mr. Whitney is well fitted for this work through his experience as a field engineer with the National Board of Fire Underwriters from 1912 to 1925. In this capacity he came to know municipal and fire prevention authorities in many cities throughout the country and achieved marked success in drafting insurance regulations which eliminated many fire hazards.

Kaiser New President of Consulting Engineers Association

At the January meeting of the New York Association of Consulting Engineers, held at the Transportation Club in the Biltmore Hotel, January 5, the following officers were elected for 1931: Chris Kaiser of Kaiser, Muller and Davies, president. Mr. Kaiser succeeds Clyde R. Place; Otto E. Goldschmidt was elected vice-president, and Alfred L. Jaros was reelected secretary.

This association meets monthly and is engaged in carrying on work in connection with the professional problems of its members.

French Addresses Western New York Chapter on Unit Heaters

Donald French, president of the Industrial Unit Heater Association, addressed the monthly meeting of the Western New York Chapter at Buffalo, December 8. Mr. French outlined the provisions of the test code for unit heaters, which has been worked out in conjunction with the A.S.H.V.E.

News of the Month

Coal Production Decreased in 1930

A decrease of sixty-four million tons of coal is reported for the first eleven months of 1930, as compared with the same period in 1929.

The greater part of the heavy decline in the consumption of coal during 1930 may be attributed to the industrial depression. Aside from this factor, there are others of a permanent nature which are assuming increasing importance in their bearing on the future of the coal market. These other factors may be divided into two groups, external influences, and those within the industry itself.

Among the external factors involved are (1) greater efficiency in the burning of coal by the larger consumers, such as public utilities and industrial plants; (2) competition of fuel oil and natural gas; (3) efforts of the Canadian government to substitute coke for anthracite, thus shifting a part of the coal market from the anthracite to the bituminous operators; (4) the development of water power, particularly in Canada, and (5) increased production of coal in foreign countries.

Cornstalk Insulation

The Bureau of Standards has published a 27-page booklet covering the production and properties of cornstalk insulation. Of most interest to the heating and ventilating engineer is the final product. The main properties are summed up in Table I, here reproduced. The average weight was 645 lb. per 1000 sq. ft. or about 16 lb. per cu. ft. The only conductivity reported was 0.39, with the temperature of the test unstated. Drying reduced the coefficient to 0.36.

Steel Heating Boiler Institute Issues Revised Rating Code

An important move in the direction of a uniform method of selecting steel heating boilers is amended by the Steel Heating Boiler Institute in the form of a revised sheet, "Method of Boiler Selection," announces as of November 12, 1930, cancelling and superseding the sheet of May 21, 1930. Whereas previously the factors covering the estimated heat emission of piping, connecting radiation and other apparatus to the boiler were designated for each 1000 sq. ft. of equivalent radiation, in the new sheet these factors are given for each 50 sq. ft. beginning with a net steam load of 1000 sq. ft.

Where the additional factors give greater accuracy in the selection of steel heating boilers the significance of the revision lies in the fact that it brings the institute's method of boiler selection in harmony with that adopted by the Heating and Piping Contractors National Association in its recently published boiler ratings.

Tests made by the institute determined the factors and developed the fact that heat emission from piping and connecting radiation reaches a maximum of 30% at 1000 sq. ft. and drops steadily as the steam load increases, reaching 25.6% at 5000 sq. ft. and 20% at 1000 sq. ft.

Better Business Bureau Issues Specifications for Dwellings

The Better Business Bureau of St. Louis has published a 95-page booklet on "Good Practice Specifications for Building Construction," for the use of the people of that city interested in the building or buying of a home.

The specifications are the result of a cooperative work among twenty-two councillors who included architects, engineers, contractors, and the various industries represented and interested in the construction of dwellings. E. B. Langenberg, president of the Langenberg Manufacturing Company, represented the heating and ventilating industry.

The specifications recommend that if a warm-air furnace heating system is to be installed, it shall be installed according to the latest edition of the Standard Code of the National Warm Air Heating Association; the chimney size shall comply with the smoke regulations of St. Louis, and shall extend three feet above a flat roof or two feet above the ridge of a peak roof. It shall have no other connection to it.

Boiler sizes for water, vapor, and steam heating systems shall be of the size recommended by the code of the Heating and Piping Contractors' National Association. It is recommended that the piping shall conform with practice and sizes recommended by the A.S.H.V.E. and that the radiation be calculated by the Society's Heat Loss Tables as published in the *Guide*.

Insulation, where used, shall have a thermal conductivity of not more than 0.35 B.t.u. per in. of thickness per sq. ft. of area, per hour, per degree Fahrenheit temperature difference. It is further recommended that not less than one inch fibrous composition board, quilt or blanket, or not less than three inches of flaky or fibrous mineral substance be applied as insulation against heat losses through the roof.

The specifications cover all phases of dwelling construction. They are not presented as a substitute for recognized procedure in building a home, that is, the engaging of a competent architect, but for use in such cases where architects are not consulted.

The book was prepared by the Construction Industries Division of the Better Business Bureau of St. Louis.

Beating treatment		Sizing (per cent by weight of dry fiber)			Properties of pulp			Water in sheet		Properties of board		
Duration of beating	Duration of washing	Rosin size	Alum	Glue	pH	Consistency	Freeness	As it leaves mat-forming machine—	As it leaves board-forming machine—	Transverse strength	Thickness	Weight per 1,000 (pounds sq. ft. ½-inch thick)
Hours	*Hours*					*Per cent*	*Seconds*	*Per cent*	*Per cent*	*Lbs./in.*	*Inches*	
6	.....	....	.....	.....	7.00	2.02	.....	72.6	70.2	.....	....	.....
.....	0.5	....	.....	.....	7.00	2.00	346	.....	.....	.....	5/16	.....
13	6.0	3	4.5	1	4.80	2.22	284	84.1	81.4	247	17/32	495
13	3.0	3	4.5	1	4.95	2.09	1,000	84.5	72.6	225	½	525
7	3.0	4	5.0	.....	4.50	.....	.....	87.2	68.7	225	½	635
5	3.5	2	4.0	.....	5.10	2.49	409	88.2	76.7	170	5/8	600
5	3.0	2	5.0	.....	4.60	1.80	536	85.8	73.9	235	½	580
4.5	.....	2	5.0	.....	4.90	2.38	.....	87.7	72.0	234	7/16	635
4	.....	2	5.0	.....	4.60	2.87	1,000	87.1	73.7	236	13/32	595
4	.....	2	5.0	.....	4.60	2.27	448	86.5	74.4	232	½	578
3.25	.....	2	5.0	.....	4.60	2.29	511	86.6	75.9	200	5/8	552

Table 1, from "Manufacture of Insulating Board from Cornstalk"

News of the Month

District Heating Developments in Winnipeg

We are informed by J. G. Glassco, of the City of Winnipeg Hydro-Electric System, that the recent report in the Canadian press, reprinted in some American newspaper, to the effect that the whole city of Winnipeg would soon enjoy a complete district heating service is erroneous. Mr. Glassco states that while it is true that two private interests have extended such a service to certain residential districts and that the city-owned utility has been authorized to extend its lines, it is not expected that the extension will be city-wide. It is planned, however, to make the service available in certain sections. As heretofore, the city-owned service will be linked with the adjacent hydro plants which have been extended recently.

Cincinnati Association of Allied Solid Fuel Heating Industries Formed

An association to support the National Committee of Ten movement in the coal and heating industries was formed in Cincinnati, December 4, under the name of the Cincinnati Association of Allied Solid Fuel Heating Industries. The purpose of the organization is to provide the public with a central organization where heating contingencies can be handled as a single problem.

Among the speakers were W. L. Mayne of the United States Radiator Corporation, who served as temporary chairman, Lorin W. Smith, secretary of the National Committee of Ten, Homer R. Linn, American Radiator Corporation, Harvey Manny, representing the National Coal Association, and W. R. Judd.

Business Conditions Discussed at Michigan Chapter

E. R. Shaw, Michigan representative of the United States Chamber of Commerce, was the principal speaker of the evening at the monthly meeting of the Michigan Chapter, held December 15, at the Cadillac Athletic Club, Detroit. Ninety members and guests attended.

Among the interesting statements made by Mr. Shaw was one to the effect that stock and bondholders have increased thirty-fold from 1909 to 1929. This has a direct effect on business conditions today, inasmuch as a great percentage of these stock and bondholders flooded the market when the crash came in 1929.

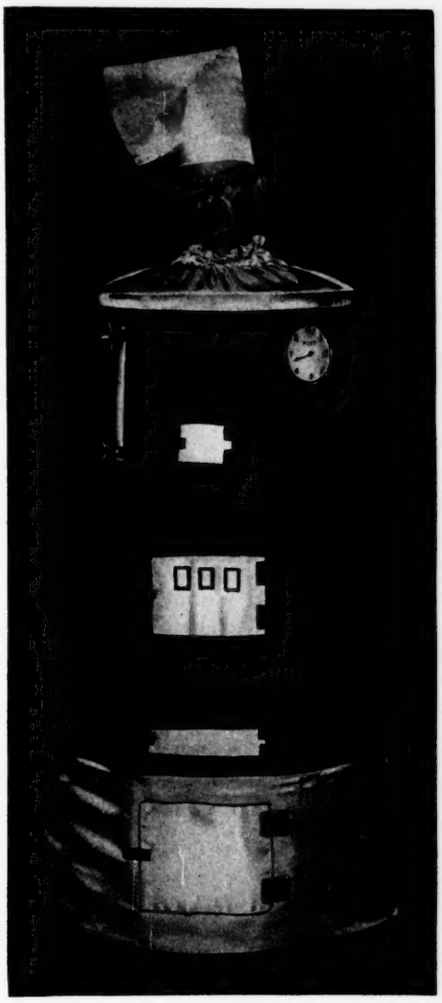
Mr. Shaw predicted that it would be many years before business conditions would reach the 1929 average, but that by the middle of 1931 business conditions will equal the average from 1925 to 1929. He indicated that the Chamber of Commerce survey showed definitely that unnecessary restraint in buying of necessities is largely responsible for the delay in getting business back to normal.

Thomas P. Chester of Pittsburgh extended a cordial invitation to the chapter members to attend the annual meeting of the Society to be held in Pittsburgh in January.

After the regular business meeting, the annual Keno party was held.

Minneapolis-Honeywell May Buy Time-O-Stat

Minneapolis-Honeywell Regulator Company, manufacturers of temperature control devices, has arranged to acquire the Time-O-Stat Controls Co., Elkhart, Ind., subject to approval of Time-O-Stat's stockholders, according to an announcement made January 28.



International Newsreel Photo

Arthur J. Arwine in a Costume Representing a Heating Boiler Worn at the Annual Beaux Arts Ball in New York, January 23

Minneapolis-Honeywell Regulator Company is the result of a merger in the fall of 1927 of the Minneapolis Heat Regulator Company and the Honeywell Heating Specialties Company of Wabash, Ind. The Minneapolis company is a descendant of the Consolidated Temperature Controlling Company, which later became the Electric Thermostat Company, and reorganized in 1893 as the Electric Heat Regulator Company. In 1912 the name was changed to the Minneapolis Heat Regulator Company. The Honeywell Heating Specialties Company was organized in 1906 and incorporated in 1916.

Time-O-Stat Controls Company was formed in January, 1929, by a merger of the Leachwood Co., Janesville, Wis., The Time-O-Stat Corp., Milwaukee, the Absolute Con-Tac-Tor Corp., Elkhart, Ind., and the Cramblet Engineering Corp., Milwaukee.

Julius K. Luthe, president of Time-O-Stat, and Neil B. Dawes, of Lane, Rolo-son & Co., Chicago, will become directors of Minneapolis-Honeywell.

Oil Burners and School Equipment Discussed by Detroit Engineers

At the regular monthly meeting of the Michigan Chapter of the A.S.H.V.E., January 12, Ray S. M. Wilde, consulting engineer of Detroit, gave a description of the mechanical equipment used in the Cranbrook School for Boys, located in Bloomfield, Ill., designed by Prof. Eliel Saarinen and of which Mr. Wilde was the engineer. The speaker's talk covered the reasons for the particular design and included a discussion of the results which have been obtained.

Philip S. Russel, resident engineer of the Detroit Lubricator Company, addressed the audience on the subject of "Oil and the Requirements Necessary for its Proper Combustion." One point covered, which excited considerable interest, was that low stack temperatures of oil-fired boilers did not necessarily mean a high overall efficiency of boilers, but, to the contrary, quite often indicated excess air. He also stated that the color of oil in combustion was always clear and any other color of the flame which may exist is simply due to the carbon in the flame.

A. G. Hann, field engineer of the Detroit Lubricator Company, presented a discussion on the "Developments in Practice and Devices because of the Use of Oil as a Fuel."

The meeting closed with the showing of a motion picture, "Carbon Monoxide —The Unseen Danger," a reel released by the Bureau of Mines at Pittsburgh.

Eighty-one members and guests were present.

Presidents and Secretaries Discuss Regional Management Conferences

The value of regional management conferences as an educational medium for the contractor, their effect on vitalizing association work and increasing membership, and the follow-up work for the conferences were the central topics for discussion at the ninth annual Presidents' and Secretaries' Conference, held under the auspices of the Plumbing and Heating Industries Bureau, at the Congress Hotel, Chicago, January 5-7.

One entire day was devoted to the presentation of a typical regional management conference. Nearly all of the following day was devoted to a frank discussion of the conferences and the follow-up work necessary in order to insure the lasting effects of the educational work begun at the conference.

Every secretary and president who attended the meeting of association officials and who saw the sales demonstrations went away with the definite opinion that at last the heating and plumbing industries have found a method of contractor education that is sure to result in the more general adoption of improved methods of merchandising and up-to-date methods of business management.

Building Decreases but Radiator Heating Increases in Cleveland

B. Y. Kinzey, secretary of the Heating and Piping Contractors Cleveland Association, stated at the recent annual banquet of that organization that while there was a decrease of 35.6% in the actual number of residences built in Greater Cleveland during the first 11 months of 1930, radiator heating was specified for 56 more jobs than in the same period of 1929. Mr. Kinzey stated that in 1929 it was only above $10,000 that radiator heating was specified in a majority of jobs. However, in 1930 radiator heating was specified in a majority of the jobs in the $7,500-$10,000 class as well. In addition, radiator heating made advances in the lower price groups.

At the business meeting preceding the banquet, the following officers and directors were elected for the coming year: President, Robert J. Mayer; vice-president, Edward D. Maurer; secretary, Adolph Weiss; treasurer, Walter Rhoton, and directors, E. L. Heinle, W. B. Van Sickle, Ross Brewer, A. C. Banko and Gerge Spohn.

A.O.B.A. Votes Increased Appropriation for Research

At the quarterly meeting of the directors of the American Oil Burner Association, held at Philadelphia, January 13, an increased appropriation for the technical research program now being carried on at Yale University by Prof. L. E. Seeley was approved. Amendments to the model ordinance, as recommended by the committee on uniform ordinances, were adopted by the board.

The slogan of the Oil Heating Institute was changed from "Oil Heat as Benevolent as Sunshine" to "Burn Oil, the Modern Fuel," to conform with the slogan of the A.O.B.A.

A resolution was adopted endorsing the Second International Heating and Ventilating Show to be held in Cleveland in 1932 and recommending it to the membership of the association.

Membership applications of the Chase Brass and Copper Co., Inc., Waterbury, Conn., and Engineering Publications, Inc., of Chicago, were accepted.

Those attending the meeting were W. F. Tant, Haldeman Finnie, W. C. McTarnahan, W. J. Smith, J. H. Hirsch, E. P. Bailey, John H. Blake, Jr., C. E. Campbell, Albert Fleischmann, G. K. Cole, J. F. Griswold, W. M. C. Kimber, John H. McIlvaine, R. M. Sherman, and Harry F. Tapp.

Gas-Fired Central Heating Installations Increase 20%

THE total number of gas-fired central heating installations, reported by 176 manufactured gas companies, grew from 39,955, as of July 1, 1929, to 47,857, on July 1, 1930, an increase of approximately 20%, according to the report of the Committee on House Heating of the American Gas Association, presented at the recent convention held in Atlantic City.

The total number of central heating installations reported by the 31 natural gas companies was 40,712 on July 1, 1929, and 66,664 on July 1, 1930, an increase of 64%. Combining the reports from the 176 manufactured and the 31 natural gas companies gives the total number of central heating installations connected to the lines of these companies on July 1, 1929, as 80,667 as compared with a total of 114,521 on July 1, 1930, an increase of approximately 41%.

The relative increase shown in the manufactured and natural gas territories served by these companies was effected by the fact that several of the larger companies changed over to natural gas during the period and were thereby able to achieve very gratifying increases in the number of customers using gas-fired central heating equipment.

The rapid growth in the installation of conversion burners in manufactured gas territory is indicated by the fact that conversion equipment constituted 19% of the total installations on July 1, 1930.

As regards type of heating systems using manufactured gas as a fuel, the most marked increase occurred in the steam, hot water, and vapor jobs which showed a growth of 33% during the year; while for the warm air systems the figure was 8%. On July 1, 1929, steam, hot water, and vapor installations comprised 48% of the total manufactured gas installations, whereas by July 1, 1930, this figure had grown to 53%.

In reviewing the returns from natural gas companies it should be borne in mind that tabulations show only the gas-fired central heating plant installations and do not cover homes heated with gas which employ any form of individual room heaters—a practice quite common in natural gas territory.

While the 31 natural gas companies reporting represent a relatively smaller proportion of this branch of the industry than was true of the manufactured gas reports, it is interesting to compare the trends in these two branches of the industry.

The growth in the number of conversion burner installations reported by the natural gas companies was 80% during the year, whereas the number of straight gas designed installations increased only 41%. Whereas, in the manufactured field, conversion burners represented only 19% of the total installations, in the natural gas field conversion burners represented 57% of the total installations on July 1, 1929, and by the following year this proportion had risen to 63%.

The growth during the year in number of warm air systems, both conversion and straight gas designed, was 66% as compared with an increase of 58% in steam, hot water and vapor installations. On July 1, 1930, warm-air installations comprised 77% of the total installations reported by the 31 natural gas companies.

The complete tabulation of data on gas-fired central house-heating installations, from the American Gas Association report, is given on the pages which follow. The data include both manufactured and natural gas systems.

DATA ON GAS FIRED CENTRAL HOUSE HEATING INSTALLATIONS REPORTED BY 176 MANUFACTURED GAS COMPANIES FOR YEARS ENDED JULY 1, 1929, AND JULY 1, 1930

Company	Total Gas Fired Central Heating Installations July 1			Straight Gas Designed Jobs: Steam, Hot Water and Vapor Installations			Straight Gas Designed Jobs: Warm Air Installations			Conversion Jobs: Steam, Hot Water and Vapor Installations			Conversion Jobs: Warm Air Installations			Estimated Annual Gas Consumption (MCF) of Central Heating Installations		
	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.
Alabama																		
Tri-City Gas Company	2	0	2	0	0	0	0	0	0	2	0	2	0	0	0	300	0	300
Birmingham Gas Company	700	0	700	112	0	112	168	0	168	168	0	168	252	0	252	237,000	0	237,000
Arizona																		
Tucson Gas, Electric Light & Power Company	24	2	22	21	1	20	3	1	2	0	0	0	0	0	0	6,000	500	5,500
California																		
Madera Gas Co.	7	5	2	5	3	2	2	2	0	0	0	0	0	0	0	1,086	775	311
San Diego Consolidated Gas and Electric Co.	1,251	1,060	191	54	44	10	1,181	1,000	181	16	16	0	0	0	0	120,000	111,000	9,000
Connecticut																		
Danbury & Pethel Gas and Electric Co.	25	22	3	19	16	3	2	2	0	4	4	0	0	0	0	23,108	22,860	248
New Britain Gas Light Company	3	2	1	2	2	0	0	0	0	1	0	1	0	0	0	1,717	1,369	348
City of Norwich Gas & Electric Dep't	11	9	2	5	5	0	1	1	0	5	3	2	0	0	0	7,033	7,799	—766
Derby Gas and Electric Company	28	20	8	16	10	6	8	8	0	3	2	1	1	0	1	16,016	13,158	2,858
New Haven Gas Light Company	422	243	179	316	180	136	93	59	34	11	3	8	2	1	1	174,000	80,000	94,000
Greenwich Gas Company	97	42	55	87	39	48	3	0	3	7	3	4	0	0	0	50,632	20,400	30,232
Stamford Gas & Electric Company	137	108	29	132	108	24	1	0	1	4	0	4	0	0	0	84,408	51,982	32,426
Connecticut Light and Power Co: Waterbury Gas Light Company	202	121	81	142	92	50	47	24	23	10	4	6	3	1	2	90,236	52,750	37,486
Winsted Gas Company	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	200	0	200
Connecticut Power Company	125	83	42	98	79	19	3	3	0	22	0	22	2	1	1	52,116	37,345	14,771
Delaware																		
Dover Gas Light Company	4	1	3	4	1	3	0	0	0	0	0	0	0	0	0	818	200	618
Wilmington Gas Company	67	50	17	61	45	16	6	5	1	0	0	0	0	0	0	51,871	36,390	15,481
District of Columbia																		
Washington Gas Light Company / Georgetown Gas Light Company	512	419	93	485	400	85	23	19	4	4	0	4	0	0	0	122,571	85,000	37,571
Florida																		
Florida Public Service Company	4	0	4	0	0	0	2	0	2	1	0	1	1	0	1	135	0	135
Georgia																		
Georgia Power Company	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	180	180	0
Idaho																		
Northwest Cities Gas Company	34	4	30	4	4	0	30	0	30	0	0	0	0	0	0	2,000	800	1,200
Illinois																		
Peoples Power Company	31	23	8	11	9	2	19	13	6	0	0	0	1	1	0	7,353	6,470	883
Illinois Power Company	30	26	4	23	19	4	7	7	0	0	0	0	0	0	0	18,043	15,704	2,339
Princeton Gas Company	11	1	10	4	1	3	0	0	0	1	0	1	6	0	6	1,346	800	546
Illinois Public Utility (Freeport)	62	38	24	20	11	9	38	27	11	4	0	4	0	0	0	28,141	13,574	14,567
Public Service Co. of No. Illinois	2,432	2,081	351	2,073	1,833	240	235	188	47	122	58	64	2	2	0	1,094,400	936,450	157,950
Central Illinois Light Company	101	28	23	34	27	7	66	0	66	1	1	0	0	0	0	38,260	33,259	5,001
Illinois Power Company (Springfield)	30	26	4	23	19	4	7	7	0	0	0	0	0	0	0	18,043	15,704	2,339
Peoples Gas Light & Coke Company	2,969	2,477	492	1,858	1,712	146	716	640	76	183	85	103	212	45	167	1,416,557	1,144,936	271,621
Rockford Gas Light & Coke Company	101	18	83	48	13	35	36	5	31	7	0	7	10	0	10	34,201	9,341	24,860
Illinois Northern Utilities Company	2	0	2	1	0	1	1	0	1	0	0	0	0	0	0	943	0	943
Union Gas and Electric Co.	27	19	8	6	5	1	17	14	3	1	0	1	3	0	3	10,742	9,322	1,420
Illinois Public Utility Company	14	10	4	4	4	0	10	6	4	0	0	0	0	0	0	45,915	43,040	2,875
North Shore Gas Company	358	293	65	287	245	42	47	36	11	19	10	9	5	2	3	254,804	203,222	51,582
Alton Gas Company	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Central Illinois Public Service Co.	229	145	84	105	78	27	121	67	54	0	0	0	3	0	3	84,000	48,600	35,400
Western United Gas & Electric Co.	452	322	130	314	231	83	132	86	46	2	2	0	4	3	1	227,000	198,000	29,000
Indiana																		
Citizens Gas Company	70	38	32	13	8	5	37	18	19	2	2	0	18	10	8	21,000	15,000	6,000
Southern Indiana Gas & Electric Co.	24	17	7	11	9	2	13	8	5	0	0	0	0	0	0	10,156	7,423	2,733
Indiana-Ohio Public Service Company	1	1	0	0	1	1	0	0	0	1	1	0	0	0	0	320	835	—515
Iowa																		
Ottumwa Gas Company	1	1	0	0	0	0	1	1	0	0	0	0	0	0	0	400	480	—80
Iowa Electric Company	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	475	475	0
Iowa Southern Utilities Company	2	0	2	2	0	2	0	0	0	0	0	0	0	0	0	1,200	0	1,200
Creston Gas & Electric Company	4	1	3	3	1	2	1	0	1	0	0	0	0	0	0	1,400	200	1,200
Peoples Light Company	24	11	13	8	6	2	16	5	11	0	0	0	0	0	0	10,292	5,153	5,139
Peoples Gas & Electric Company	3	1	2	3	1	2	0	0	0	0	0	0	0	0	0	1,350	400	950
Sioux City Gas and Electric Company	26	7	19	10	6	4	15	1	14	0	0	0	1	0	1	10,400	3,500	6,900
Keokuk Electric Company	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Iowa Railway and Light Corp.	7	4	3	5	3	2	2	1	1	0	0	0	0	0	0	6,250	4,300	1,950
Maine																		
Citizens Gas Company	2	0	2	0	0	0	0	0	0	2	0	2	0	0	0	355	0	355
Lewiston Gas Light Company	29	25	4	21	22	—1	5	3	2	2	0	2	1	0	1	13,221	9,770	3,451
Cumberland County Power and Light	8	5	3	8	5	3	0	0	0	0	0	0	0	0	0	4,169	3,196	973
Portland Gas Light Company	158	134	24	142	123	19	11	11	0	5	0	5	0	0	0	86,390	80,831	5,559
Maryland																		
Consolidated Gas Elec. Lt. & Pr. Co.	2,626	1,984	642	2,415	1,794	621	107	86	21	91	93	—2	13	11	2	1,052,210	950,000	102,210
Hagerstown Light and Heat Company	12	4	8	11	4	7	1	0	1	0	0	0	0	0	0	4,893	652	4,241
Crisfield Light & Power Company	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	423	0	423
Washington Suburban Gas Company	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	419	0	419
Massachusetts																		
Gas Light North Attleboro Company	2	1	1	2	1	1	0	0	0	0	0	0	0	0	0	1,000	300	700
Athol Gas & Electric Company	3	3	0	2	2	0	1	1	0	0	0	0	0	0	0	1,050	1,050	0
Gardner Gas Fuel & Light Co.	4	3	1	3	3	0	0	0	0	1	0	1	0	0	0	1,400	1,050	350
Lowell Gas Light Co.	193	146	47	98	88	10	9	8	1	76	49	27	10	1	9	93,980	48,023	45,957
Boston Consolidated Gas Company	1,661	1,234	427	1,239	915	324	265	184	81	132	121	11	25	14	11	1,000,000	750,000	250,000
Charlestown Gas and Electric Co.	11	6	5	8	5	3	1	1	0	1	0	1	1	0	1	5,000	2,600	2,400
Haverhill Gas Light Company	343	278	65	220	216	4	31	27	4	78	31	47	14	4	10	151,900	116,255	35,645
Fall River Gas Works Co.	252	189	63	150	138	12	17	16	1	81	35	46	4	0	4	101,249	59,778	41,474
Salem Gas Light Company	10	3	7	7	2	5	1	0	1	2	1	1	0	0	0	4,565	1,371	3,194
Haverhill Electric Company	3	2	1	0	0	0	3	2	1	0	0	0	0	0	0	1,180	930	250
Lexington Gas Company	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	373	0	373
Malden & Melrose Gas Light Co.	154	119	35	97	85	12	25	22	3	31	11	20	1	1	0	58,692	48,353	10,339
Beverly Gas & Electric Company	7	4	3	2	2	0	4	2	2	1	0	1	0	0	0	2,526	612	1,914
Fitchburg Gas & Electric Light Co.	29	23	6	26	21	5	3	2	1	0	0	0	0	0	0	12,148	9,590	2,558
Suburban Gas & Electric Company	17	11	6	12	8	4	2	2	0	3	1	2	0	0	0	8,971	5,805	3,166
Springfield Gas Light Company	137	121	16	117	107	10	16	12	4	4	2	2	0	0	0	82,000	72,000	10,000
Pittsfield Coal Gas Company	40	28	12	39	27	12	1	1	0	0	0	0	0	0	0	22,588	15,862	6,726
Brockton Gas Light Company	143	127	16	127	120	7	6	1	5	10	6	4	0	0	0	77,990	65,973	12,017
North Adams Gas Light Company	7	4	3	6	3	3	0	0	0	1	1	0	0	0	0	2,030	700	1,330
Plymouth Gas Light Company	8	8	0	6	6	0	2	2	0	0	0	0	0	0	0	1,774	1,690	84
New England Gas & Electric System	845	518	327	610	423	187	0	0	—	235	95	140	0	0	0	321,100	129,500	191,600
Leominster Gas Light Company	5	0	5	5	0	5	0	0	0	0	0	0	0	0	0	2,000	0	2,000
Michigan																		
Iron Mountain Gas Company	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	450	450	0
Holland Gas Company	3	0	3	2	1	0	1	0	1	0	0	0	0	0	0	1,300	430	870

DATA ON GAS FIRED CENTRAL HOUSE HEATING INSTALLATIONS REPORTED BY 176 MANUFACTURED GAS COMPANIES FOR YEARS ENDED JULY 1, 1929, AND JULY 1, 1930

Company	Total Gas Fired Central Heating Installations July 1			Straight Gas Designed Jobs: Steam, Hot Water and Vapor Installations			Straight Gas Designed Jobs: Warm Air Installations			Conversion Jobs: Steam, Hot Water and Vapor Installations			Conversion Jobs: Warm Air Installations			Estimated Annual Gas Consumption (MCF) of Central Heating Installations		
	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.
Battle Creek Gas Company	170	163	7	75	71	4	93	90	3	1	1	0	1	1	0	81,503	83,890	—2,387
Grand Rapids Gas Light Company	39	25	14	17	12	5	20	12	8	0	0	0	2	1	1	20,883	12,563	8,320
Detroit City Gas Company	641	568	73	522	476	46	66	43	23	40	30	10	13	19	—6	473,497	359,576	113,921
Consumers Power Company	508	378	130	290	223	67	213	152	61	4	2	2	1	1	0	291,427	204,289	87,138
National Utilities Co. of Mich. (Monroe)	4	0	4	4	0	4	0	0	0	0	0	0	0	0	0	1,850	0	1,850
National Utilities Co. of Mich. (Grand Haven)	2	2	0	2	2	0	2	2	0	0	0	0	0	0	0	145	897	—752
Michigan Federated Utilities (Mt. Clemens)	17	12	5	13	11	2	2	1	1	0	0	0	2	0	2	10,215	9,988	227
Michigan Federated Utilities (Sault Ste. Marie)	9	6	3	6	4	2	2	2	0	0	0	0	1	0	1	6,000	3,600	2,400
Minnesota																		
Northern States Power Co., St. Paul	91	89	2	84	82	2	0	0	0	2	2	0	5	5	0	30,401	30,934	—533
Minneapolis Gas Light Co.	257	212	45	222	181	41	14	10	4	15	15	0	6	6	0	135,000	110,000	25,000
Missouri																		
St. Louis County Gas Company	596	448	148	526	391	135	64	52	12	4	3	1	2	2	0	300,000	217,000	83,000
Laclede Gas Light Company	513	467	46	318	359	—41	69	95	—26	95	11	84	31	2	29	281,005	248,445	32,560
Nebraska																		
Metropolitan Utilities Dist.	337	194	143	61	27	—34	267	161	106	7	5	2	2	1	1	108,189	60,157	48,032
Northwestern Public Service Company	6	0	6	1	0	1	5	0	5	0	0	0	0	0	0	3,284	0	3,284
Central Power Co., (Hastings only)	3	3	0	2	2	0	1	1	0	0	0	0	0	0	0	950	950	0
Nevada																		
Sierra Pacific Power Co.	5	5	0	1	1	0	4	4	0	1	1	0	0	0	0	1,300	1,110	190
New Hampshire																		
Portsmouth Gas Co.	22	15	7	17	11	6	5	4	1	0	0	0	0	0	0	7,700	5,250	2,450
Concord Gas Company	51	48	3	51	48	3	0	0	0	0	0	0	0	0	0	2,500	2,400	100
Public Service Co. of New Hampshire	67	55	12	55	45	10	12	10	2	0	0	0	0	0	0	33,230	27,230	6,000
Manchester Gas Co.	46	36	10	42	35	7	4	1	3	0	0	0	0	0	0	27,744	25,288	2,456
New Jersey																		
Salem Gas Light Co.	7	1	6	6	1	5	1	0	1	0	0	0	0	0	0	1,403	295	1,108
County Gas Company	51	34	17	22	14	8	3	0	3	25	19	6	1	1	0	22,000	16,000	6,000
Atlantic City Gas Co.	262	192	70	249	182	67	1	0	1	12	10	2	0	0	0	139,305	84,920	54,385
Jersey Central Power & Light Co.	405	311	94	389	303	86	7	3	4	6	1	5	3	4	—1	135,642	97,348	38,294
Public Service Electric & Gas Co.	1,130	704	426	1,001	635	366	95	45	50	24	22	2	10	2	8	506,666	314,815	191,851
New Jersey Power & Light Co.	6	4	2	6	4	2	0	0	0	0	0	0	0	0	0	3,200	2,100	1,100
New York																		
Kings County Lighting Co.	260	159	101	242	151	91	5	4	1	11	3	8	2	1	1	117,327	73,600	43,727
Brooklyn Boro Gas Co.	102	60	42	99	60	39	0	0	0	3	0	3	0	0	0	39,633	13,750	25,883
Utica Gas and Electric Company	231	169	62	110	96	14	94	66	28	15	4	11	12	3	9	135,483	116,434	19,049
New York State Electric & Gas Corp. N. & N. E. Div.	8	7	1	6	6	0	1	1	0	1	0	1	0	0	0	3,200	2,800	400
New York & Richmond Gas Co.	78	33	45	60	29	31	7	1	6	9	2	7	2	1	1	29,000	12,000	17,000
Brooklyn Union Gas Co.	556	434	122	527	420	107	7	4	3	18	8	10	4	2	2	219,900	173,600	46,300
Empire Gas & Electric Co.	8	4	4	5	2	3	3	2	1	0	0	0	0	0	0	5,571	2,201	3,370
New York State Electric & Gas Corp. (Oneonta Div.)	44	5	39	26	4	22	4	1	3	6	0	6	8	0	8	15,500	1,500	14,000
New York State Electric & Gas Co. (Lockport)	26	11	15	6	1	5	7	1	6	4	3	1	9	6	3	14,445	5,513	8,932
Northern New York Utilities	2	2	0	0	0	0	2	2	0	0	0	0	0	0	0	2,400	0	2,400
Rockland Light & Power Co.	21	18	3	16	14	2	4	4	0	1	0	1	0	0	0	8,100	6,444	1,656
Consolidated Gas Co. of New York & Affl. Cos.	1,641	1,119	522	1,326	975	351	97	56	41	204	83	121	14	5	9	987,000	867,784	119,216
Syracuse Lighting Co.	157	42	115	44	10	34	31	30	1	2	0	2	80	2	78	55,000	20,000	35,000
Rochester Gas & Electric Corp.	577	433	144	291	215	76	263	205	58	10	5	5	13	8	5	333,395	235,438	97,957
North Carolina																		
Piedmont Utilities Co. Inc.	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	408	0	408
*Tide Water Power Co.	1	1	0	0	0	0	0	0	0	1	1	0	0	0	0	1,320	1,320	0
Asheville Gas Company	3	0	3	3	0	3	0	0	0	0	0	0	0	0	0	900	0	900
Ohio																		
West Ohio Gas Company	35	27	8	11	10	1	11	8	3	7	5	2	6	4	2	17,000	10,000	7,000
Western Ohio Public Service Co.	6	2	4	2	0	2	1	1	0	2	0	2	1	1	0	1,300	515	785
Oregon																		
Northwest Cities Gas Co.	25	18	7	4	4	0	2	2	0	19	12	7	0	0	0	2,340	1,920	420
†Portland Gas & Coke Co.	16,508	17,168	—660	592	596	—4	9,609	10,135	—526	21	22	—1	6,286	6,415	—129	1,300,690	1,464,953	—164,263
Pennsylvania																		
Chambersburg Gas Company	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	327	296	31
Lebanon Valley Gas Co.	19	8	11	16	8	8	1	0	1	2	0	2	0	0	0	5,748	3,500	2,248
Waynesboro Gas Co.	11	2	9	10	2	8	1	0	1	0	0	0	0	0	0	6,262	971	5,291
Interborough Gas and Fuel Co.	4	2	2	4	2	2	0	0	0	0	0	0	0	0	0	2,457	1,208	1,249
Harrisburg Gas Company	35	19	16	31	19	12	2	0	2	1	0	1	1	0	1	16,624	11,531	5,093
Pottsville Gas Company	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	465	0	465
Philadelphia Gas Works Co.	1,700	1,240	460	1,540	1,144	396	130	84	46	24	6	18	6	6	0	600,000	434,000	166,000
Northern Liberties Gas Co.	8	6	2	3	3	0	0	0	0	0	0	0	0	0	0	1,582	1,307	275
Conewago Gas Company	7	3	4	6	3	3	1	0	1	0	0	0	0	0	0	3,574	1,379	2,195
Pennsylvania Gas & Electric Co.	35	23	12	35	23	12	0	0	0	0	0	0	0	0	0	20,457	11,008	9,449
Penn Central Light & Power Co.	12	11	1	10	9	1	2	2	0	0	0	0	0	0	0	5,000	4,000	1,000
Philadelphia Electric Co.	1,130	669	461	997	605	391	91	53	38	23	6	17	19	4	15	650,000	380,000	270,000
Pennsylvania Electric Co.	81	64	17	57	51	6	1	1	0	22	12	10	1	0	1	18,225	14,400	3,825
Metropolitan Edison Co.	23	7	16	20	7	13	3	0	3	0	0	0	0	0	0	16,000	4,000	12,000
Scranton Spring Brook Water Service Co.	39	39	0	1	1	0	0	0	0	36	36	0	2	2	0	24,394	21,878	2,516
Consumers Gas Company	75	39	36	70	35	35	5	4	1	0	0	0	0	0	0	25,958	9,221	16,737
Penn Central Light & Power Co.	9	9	0	8	8	0	1	1	0	0	0	0	0	0	0	5,010	5,010	0
Lancaster Gas Light and Fuel Co.	16	7	9	10	6	4	4	0	4	1	1	0	1	0	1	8,722	5,232	3,490
Allentown-Bethlehem Gas Co.	30	6	24	30	6	24	0	0	0	0	0	0	0	0	0	15,987	3,114	12,873
Highland Gas Company	6	1	5	6	1	5	0	0	0	0	0	0	0	0	0	2,303	101	2,199
Rhode Island																		
Blackstone Valley G. & E. Co.	159	155	4	135	141	—5	18	9	9	2	4	—2	4	1	3	75,000	63,373	11,627
Newport Gas Light Co.	11	5	6	11	5	6	0	0	0	0	0	0	0	0	0	5,000	2,500	2,500
The Narragansett Electric Co.	4	0	4	3	0	3	1	0	1	0	0	0	0	0	0	1,059	0	1,059
Pawtucket Gas Co.	313	256	57	263	234	29	27	18	9	23	4	19	0	0	0	133,395	115,916	17,479
South Dakota																		
Sioux Falls Gas Co.	21	12	9	4	1	3	15	9	6	0	0	0	2	2	0	7,700	4,200	3,500
Northwestern Public Service Co.	30	0	30	4	0	4	12	0	12	5	0	5	9	0	9	10,272	0	10,272
Tennessee																		
Washington County Gas Co.	6	5	1	5	5	0	1	0	1	0	0	0	0	0	0	2,250	2,250	0
Knoxville Gas Company	45	47	—1	20	21	—1	3	0	3	15	13	2	8	13	—5	42,321	17,722	24,599
Vermont																		
Vermont Lighting Co. (Barre)	1	0	1	1	0	1	0	0	0	0	0	0	0	0	0	679	0	679

DATA ON GAS FIRED CENTRAL HOUSE HEATING INSTALLATIONS REPORTED BY 176 MANUFACTURED GAS COMPANIES FOR YEARS ENDED JULY 1, 1929, AND JULY 1, 1930

Company	Total Gas Fired Central Heating Installations July 1			Straight Gas Designed Jobs: Steam, Hot Water and Vapor Installations			Straight Gas Designed Jobs: Warm Air Installations			Conversion Jobs: Steam, Hot Water and Vapor Installations			Conversion Jobs: Warm Air Installations			Estimated Annual Gas Consumption (MCF) of Central Heating Installations		
	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.
Virginia																		
Petersburg Gas Co.	2	2	0	2	2	0	0	0	0	0	0	0	0	0	0	446	446	0
Virginia Public Service Co.	11	9	2	10	9	1	0	0	0	1	0	1	0	0	0	3,850	3,050	800
Lynchburg Gas Co.	6	0	6	2	0	2	4	0	4	0	0	0	0	0	0	2,800	0	2,800
Consumers Utilities Co.	—	—	—	0	0	0	0	0	0	1	0	1	0	0	0	—	—	—
Department of Pub. Util., Richmond	24	6	18	22	6	16	0	0	0	2	0	2	0	0	0	20,000	8,500	11,500
Suffolk Gas & Electric Co.	2	0	2	2	0	2	0	0	0	0	0	0	0	0	0	786	0	786
Roanoke Gas Light Co.	35	10	15	21	9	12	4	1	3	0	0	0	0	0	0	10,387	5,446	4,941
Virginia Electric & Power Co.	2	2	0	2	2	0	0	0	0	0	0	0	0	0	0	1,000	1,000	0
Washington																		
Northwest Cities Gas Co.	8	1	7	0	0	0	8	1	7	0	0	0	0	0	0	480	90	390
Spokane Gas & Fuel Co.	53	41	12	27a	19b	8	26	22	4	0	0	0	0	0	0	16,158	15,527	631
Wisconsin																		
Lake Superior Dist. Power Co.	2	0	2	1	0	1	1	0	1	0	0	0	0	0	0	1,461	0	1,461
Madison Gas & Electric Co.	99	72	27	71	49	22	28	23	5	0	0	0	0	0	0	49,000	36,150	12,850
Wisconsin Valley Electric Co.	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	1,500	1,500	0
Wisconsin Power & Light Co.	64	41	23	22	16	6	42	25	17	0	0	0	0	0	0	28,443	20,965	7,478
Milwaukee Gas Light Co.	714	628	86	495	418	77	214	208	6	4	0	4	1	2	—1	332,000	292,213	39,787
Wisconsin Gas & Electric Co.	37	22	15	22	13	9	15	9	6	0	0	0	0	0	0	25,000	16,000	9,000
Wisconsin-Michigan Power Co.	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	1,157	0	1,157
Superior Water Light & Power Co.	10	10	0	8	8	0	0	0	0	1	1	0	1	1	0	4,000	4,000	0
Canada																		
Consumers Gas Co. of Toronto	46	26	20	45	25	20	0	0	0	0	0	0	1	1	0	41,000	25,000	16,000
MANUFACTURED GAS SUMMARY (176 COMPANIES)																		
Total	47,855	39,953	7,902	23,217	18,026	5,191	15,380	14,317	1,063	2,090	988	1,102	7,168	6,622	546	15,662,981	12,092,318	3,570,663
Per Cent Increase			19.8			28.8			7.4			111.5			8.3			29.5

DATA ON GAS FIRED CENTRAL HOUSE HEATING INSTALLATIONS REPORTED BY 31 NATURAL GAS COMPANIES FOR YEARS ENDED JULY 1, 1929, AND JULY 1, 1930

Company	Total Gas Fired Central Heating Installations July 1			Straight Gas Designed Jobs: Steam, Hot Water and Vapor Installations			Straight Gas Designed Jobs: Warm Air Installations			Conversion Jobs: Steam, Hot Water and Vapor Installations			Conversion Jobs: Warm Air Installations			Estimated Annual Gas Consumption (MCF) of Central Heating Installations		
	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.
Arkansas																		
Little Rock Gas & Fuel Co.	411	372	39	58	48	10	24	16	8	193	172	21	136	136	0	82.120	74,400	7,720
Arizona																		
Public Utilities Consolidated Corp.	38	38	0	0	0	0	37	37	0	0	0	0	1	1	0	2,052	2,052	0
California																		
Southern California Gas Co.	244	190	54	130	105	25	Unknown	Unknown		114	85	29	Unknown	Unknown		527,465	445,475	81,990
Colorado																		
Public Service Co. of Colorado, Denver	8,428	4,005	4,423	217	136	81	3,381	1,257	2,124	1,211	653	558	3,619	1,959	1,660	2,036,377	948,754	1,087,623
Pueblo Gas & Fuel Co.	1,083	421	662	10	4	6	216	60	156	60	54	6	435	224	211	227,322	121,200	106,122
Public Service of Colorado, Ft. Collins	1,224	1,165	59	55	50	5	135	125	10	155	150	5	285	275	10	291,644	282,120	9,524
Georgia																		
Atlanta Gas Light Co.	436	0	436	17	0	17	48	0	48	111	0	111	260	0	260	7,000	0	7,000
Indiana																		
Washington Gas and Coke Co.	23	5	18	6	3	3	3	2	1	9	0	9	5	0	5	3,407	820	2,587
Kansas																		
The Gas Service Co.	7,531	4,390	3,141	85	51	34	1,829	1,021	808	843	498	345	4,774	2,820	1,954	1,506,200	878,000	628,200
Peoples Utilities Kansas Corp.	148	91	57	4	3	1	69	50	19	23	14	9	52	24	28	31,100	19,130	11,970
The Kansas Pipe Line & Gas Co.	1,505	615	890	13	4	9	63	7	56	63	27	36	1,366	577	789	190,060	64,976	125,084
Kentucky																		
Kentucky Utilities Co. Paducah	1	1	0	1	1	0	0	0	0	0	0	0	0	0	0	769	779	—10
Louisiana																		
Baton Rouge Electric Co.	93	75	18	50	35	15	10	8	2	25	25	0	8	7	1	2,266	1,125	1,141
Southern Cities Distrib. Co.	13	13	0	2	2	0	8	8	0	1	1	0	2	2	0	2,000	2,000	0
Maryland																		
Cumberland & Allegheny Gas Co.	155	98	57	22	9	13	10	9	1	118	75	43	5	5	0	37,006	23,324	13,682
Michigan																		
Muskegon Gas Company	461	375	86	16	16	0	41	31	10	0	0	0	404	328	76	191,466	106,816	84,650
Missouri																		
Gas Service Co.	12,800	5,601	7,199	78	42	36	1,188	443	745	1,730	767	963	9,804	4,349	5,455	2,560,000	1,120,200	1,439,800
Missouri Gas and Electric Co.	1	1	0	0	0	0	1	1	0	0	0	0	0	0	0	410	410	0
New Mexico																		
Clayton Gas Company	21	21	0	1	1	0	1	1	0	8	8	0	11	11	0	6,640	6,640	0
Ohio																		
Ohio Electric Power Co.	44	42	2	15	14	1	10	10	0	13	12	1	6	6	0	4,644	4,430	214
Western Ohio Public Service Co.	6	2	4	3	1	2	1	1	0	1	0	1	1	0	1	1,712	356	1,356
Ohio Fuel Gas Company	26,400	20,300	6,100	3,500	3,000	500	10,000	9,000	1,000	2,400	1,800	600	10,500	6,500	4,000	6,600,000	5,075,000	1,525,000
Oklahoma																		
The Gas Service Co.	126	66	60	13	5	8	3	1	2	18	9	9	92	51	41	25,200	13,200	12,000
Guthrie Gas Service	30	30	0	5	5	0	3	3	0	0	0	0	22	22	0	9,000	9,000	0
Pennsylvania																		
Equitable Gas Company	1,042	558	484	808	443	365	217	115	102	10	0	10	7	0	7	312,600	167,400	145,200
South Dakota																		
Montana Dakota Power Co.	1,150	950	200	40	25	15	175	75	100	210	175	35	725	675	50	345,000	285,000	60,000
Tennessee																		
Memphis Power and Light	2,000	600	1,400	350	75	275	50	20	30	1,400	425	975	200	80	120	662,837	58,158	604,679

DATA ON GAS FIRED CENTRAL HOUSE HEATING INSTALLATIONS REPORTED BY 31 NATURAL GAS COMPANIES FOR YEARS ENDED JULY 1, 1929, AND JULY 1, 1930

Company	Total Gas Fired Central Heating Installations July 1			Straight Gas Designed Jobs: Steam, Hot Water and Vapor Installations			Straight Gas Designed Jobs: Warm Air Installations			Conversion Jobs: Steam, Hot Water and Vapor Installations			Conversion Jobs: Warm Air Installations			Estimated Annual Gas Consumption (MCF) of Central Heating Installations		
	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.	1930	1929	Incr.
Texas																		
Texas Cities Gas Co.	280	0	280	30	0	30	40	0	40	30	0	30	180	0	180	42,000	0	42,000
West Texas Utilities Co.	11	10	1	10	9	1	1	1	0	0	0	0	0	0	0	5,700	5,000	700
Texas Public Service Co.	451	234	217	6	2	*4	63	30	33	131	36	95	251	166	85	84,553	31,334	53,219
San Antonio Public Service Co.	481	421	60	7	7	0	171	156	15	158	138	20	145	120	25	82,500	72,200	10,300
West Virginia																		
Monongahela West Penn. P. S. Co.	27	22	5	14	11	3	5	4	1	4	4	0	4	3	1	410	335	75
NATURAL GAS SUMMARY (31 COMPANIES)																		
Total	56,664	40,712	25,952	5,565	4,106	1,459	17,803	12,492	5,311	9,039	5,128	3,911	33,300	18,341	14,959	16,669,940	10,584,634	6,085,305
Per Cent Increase			63.8			35.5			42.5			76.3			81.6			57.5

Sale of Conversion Burners Depends on Gas Rate

According to the report of the Sub-Committee on Conversion Burners of the American Gas Association, the sale of conversion burners has been in inverse ratio to the gas rate. The complete report follows:

A questionnaire was sent to eight manufactured gas companies and six natural gas companies, known to have been most active in the conversion field, replies to which were received from seven manufactured and five natural gas companies.

The information adduced from these reports is substantially as follows:

In the manufactured gas company's territories the operating costs of conversion burners ranged from equal to 20% higher than that for gas-designed equipment, and from 10 to 20% greater in the natural gas field; however, these figures should not be taken as conclusive as they were not based on a comparison of installations of both types of equipment in houses of similar size and construction.

There was no difference reported in the cost of servicing as between conversion and gas designed installations in either territory.

In the manufactured gas territories the sales of equipment were divided 50% gas designed and 50% conversion, while in the natural gas territory, 21% was gas designed and 79% conversion.

It is evident from these figures that the sale of conversion burners has been in inverse ratio to the gas rate, and that those companies desiring to develop the house-heating load rapidly turn to the conversion burner as the medium for accomplishing the result.

For some time the sale of the conversion unit was considered inadvisable by many companies, for the reason that they believed the use of this equipment would cause adverse criticism on gas house heating. This feeling seems to be gradually disappearing, as the use of the conversion burner becomes more widespread.

Those companies who pay particular attention to installing these units efficiently have proved that satisfactory service can be obtained by their use. In the final analysis satisfactory service should be the primary consideration, and many companies have found that the sale of the conversion burner has served a three-fold purpose—given service to the customer, increased the gas load, and lastly, increased the volume of house heating equipment sales.

This proves that the sale of the conversion job is not entirely a question of proper or improper heating service, but resolves itself, where rates are appropriate, into a question of how rapidly we wish to attach the gas house heating load.

With the conversion burners that are now available, fully equipped with all the automatic controls that are found on gas designed boilers, the sales problem has shifted from a mechanical to a sales problem.

No matter how we may personally feel towards the conversion burner it would seem that it is here to stay, and that it has a rightful place in our house heating setup.

With the advent of lower rates it is bound to become more of a factor in our sales programs and must receive due consideration.

The opinion of the committee is that both types of house-heating equipment have their own particular spheres in the sale of gas for house heating, and the installation of the conversion burner is very frequently the forerunner of the sale of a gas designed boiler.

Your committee feels that too much stress cannot be laid on the selection of the type unit used and the importance of its proper installation, with a view to efficient operation, for on this depends the success or failure of future gas loads from this type of house heating equipment.

Simplified Practice Recommendation on Pipe and Valves Reaffirmed

The division of simplified practice of the bureau of standards recently announced that simplified practice recommendation No. 57, Wrought Iron and Wrought Steel Pipe, Valves, and Fittings, has been reaffirmed by the standing committee of the industry, without change, for another year.

Reports from 12 manufacturers, in reply to a survey of their 1929 production of this product, conducted prior to the revision meeting, indicated their degree of adherence to the recommendation to be approximately 98%.

Plumbing & Heating Industries Publishes Statement of Its Accomplishments

"Facts" is the title of a recent bulletin of the Plumbing & Heating Industries Bureau, containing a brief statement of the work done by the Bureau in 1930 together with the program for 1931. In the matter of publicity the Bureau prepared newspaper articles amounting to about 473 newspaper pages reaching a circulation of over twice the population of the United States. Articles were accepted by 32 national magazines with a circulation of 4,500,000.

The Bureau states that the articles and news items were designed to interest the home owner in better plumbing and heating equipment and to emphasize the need for modernization.

In its new form of a monthly sales manual, the Bulletin published by the Bureau has appeared in ten issues, each containing sales stimulation for contractors. The booklet also states that a total of 3800 contractors and their employes have attended the 14 Regional Management Conferences which have been held since the plan was inaugurated in July, 1930.

Unit Gas Consumption Figures for Large Buildings

ALTHOUGH former reports of the Sub-Committee on Large Building Heating of the A.G.A. have dealt with straight gas-designed equipment, the increasing use of conversion burners prompted their inclusion in the 1930 report. Abstracts from the report follow:

As contrasted with previous reports, the method of comparison of this report is the total B.T.U. input per square foot of radiation per degree day. This value is termed "K." The method of comparison was changed in order to directly compare installations, regardless of the B.T.U. value of the gas. Expressing this as a formula we have:

$$K = \frac{CF \times B.T.U.}{R \times D.D.}$$

where CF equals the total cubic feet of gas consumed.

B.T.U. equals the B.T.U. content of the gas.

R equals the DCI theoretical radiation (installed radiation used where theoretical not available).

D.D. equals the degree days.

In order to afford a fair comparison all figures have been corrected on a basis of 0° F. outside and 70° F. inside temperature.

To correct a "K," obtained with 10° below zero to a standard of 0° outside and 70° inside, divide by .875 and for plus 10° outside to 70° inside divide by 1.185.

Although former reports have dealt with straight gas designed equipment only, the increasing use of conversion burners prompted their inclusion in this year's report and they afford an interesting comparison of the two types of equipment.

In comparing the results of last year with those obtained this year for steam installations the average K was 102 against 97 for this year. For hot water it was 60 against 70 for this year.

Where automatic regulation is used the majority of installations are controlled by the standard room thermostat. However, increasing use of other types of controls is noticeable. These consist chiefly of the Pendleton, described in the 1928 report, and the Powers and Johnson systems, controlling diaphragm valves on individual radiators or mains from room thermostat by means of compressed air.

The Duo-stat is an inexpensive type of control used either independently or in conjunction with a room thermostat and is particularly adaptable to large residences and apartments. It consists of two liquid filled bulbs, one of which is placed outdoors and the other on the riser of the boiler. These bulbs are connected by means of flexible tubing to a Bourdon coil, the expansion of which tips a mercury switch and, by means of electric wires, shuts off the fuel supply.

It is the total expansion of the liquid in both the outdoor bulb and the boiler bulb which determines the amount of expansion of the Bourdon coil and thereby the position of the mercury switch. When the outdoor bulb is warm, its liquid will be fairly well expanded and just a slight amount of added heat, applied to the boiler bulb, will expand the Bourdon coil to the point where it tips the mercury switch to its off position.

When the outdoor bulb is cold, its liquid will be greatly contracted and the boiler bulb will have to be heated to a high degree to expand the Bourdon coil to the point where it tips the mercury switch to its off position.

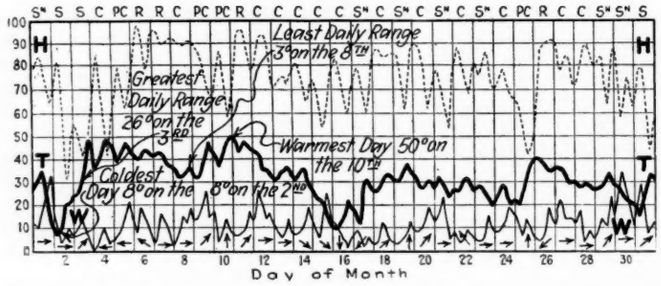
A number of other controls were investigated but the time which they have been upon the market was not sufficient enough to include them in this year's report.

It is recommended by the Committee that the association formulate a clearer definition for the degree day and that information and data pertaining to the converting of large power boilers to gas be collected and published.

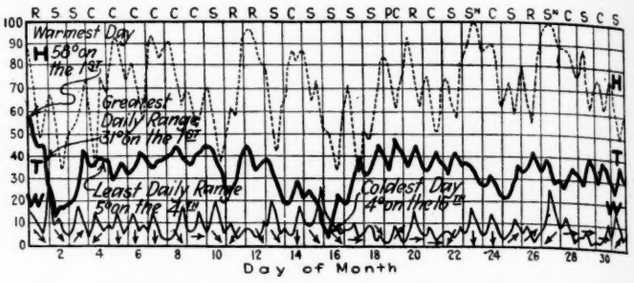
SUMMARY SHEET

No. of Install.	sq. ft. Radiation Smallest	Largest	Average	Average "K"	Remarks
		MANUFACTURED GAS			
		STEAM			
1	——	——	1466	150	Apartment
10	1047	11000	3848	92	Office
3	1425	4100	2294	77	Bank
1	——	——	4400	95	Store
2	1600	3372	2486	96	Mfg. Plant
11	702	1658	1206	109	Residence
3	1006	1840	1288	33	Church
1	——	——	4950	48	Club
1	——	——	1404	61	Library
		VACUUM VAPOR			
2	1290	1438	1364	186	Apartment
6	1475	26918	7763	83	Office
2	1750	1875	1812	105	Bank
1	——	——	5000	50	Mfg. Plant
20	727	3300	1401	105	Residence
1	——	——	6000	98	Club
1	——	——	7583	74	Garage
1	——	——	2524	126	Hospital
		HOT WATER			
1	——	——	2300	85	Office
26	1185	6100	2206	70	Residence
1	——	——	2963	38	Mfg. Plant
1	——	——	1882	93	Hospital
		NATURAL GAS			
		STEAM			
4	1030	6464	4011	141	Office
2	2588	4820	3704	113	Bank
1	——	——	1800	114	Mfg. Plant
2	1428	3453	2440	88	Church
1	——	——	1903	103	Hotel
1	——	——	1291	147	Theater
1	——	——	2022	105	Boro Hall
1	——	——	5325	132	Garage
1	——	——	4983	54	Sub Station
		VACUUM VAPOR			
6	726	1972	1288	172	Apartment
1	——	——	1610	101	Bank
1	——	——	2025	80	Church
		HOT WATER			
1	——	——	1676	70	Residence
		CONVERSION BURNERS			
		STEAM			
10	722	7200	3620	185	Apartment
5	2163	15000	8533	145	Office Bldg.
1	——	——	11000	201	Hotel
3	4200	14000	9400	165	Bank
1	——	——	8000	129	Store
1	——	——	20000	130	School
3	3600	62062	25837	62	Mfg. Plant
1	——	——	9450	73	Air Hangar
		VACUUM VAPOR			
11	2000	15000	5655	200	Apartments
		HOT WATER			
2	1341	4260	2800	72	Office Bldg.

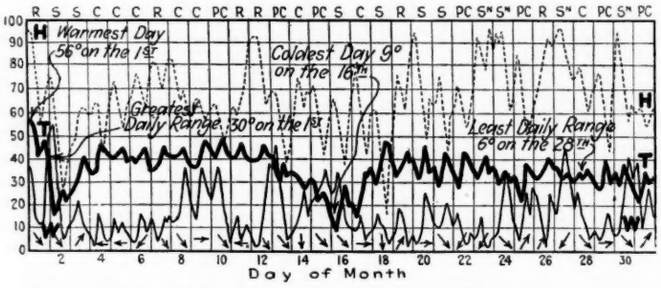
The Weather for December, 1930


Least Daily Range 3° on the 8TH
Greatest Daily Range 26° on the 3RD
Warmest Day 50° on the 10TH
Coldest Day 8° on the
3° on the 2ND
Day of Month

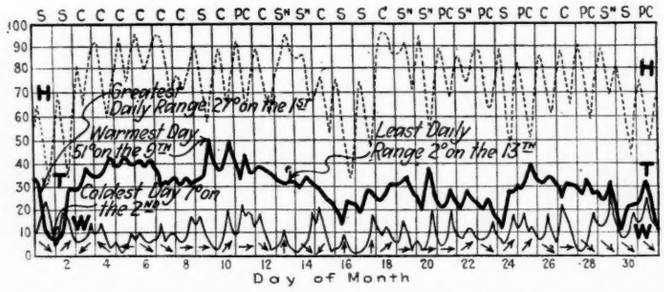
Record of the Weather in Pittsburgh for December, 1930


Warmest Day 58° on the 1ST
Greatest Daily Range 31° on the 7TH
Least Daily Range 5° on the 24TH
Coldest Day 4° on the 16TH
Day of Month

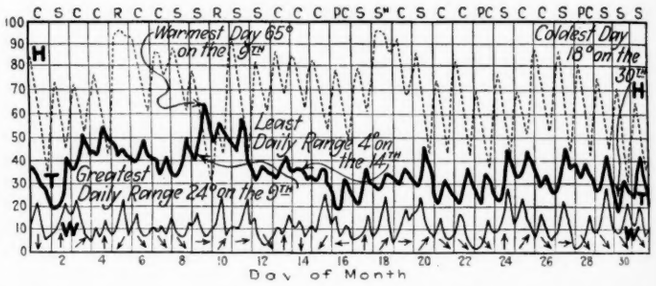
Record of the Weather in Boston for December, 1930


Warmest Day 56° on the 1ST
Coldest Day 9° on the 16TH
Greatest Daily Range 30° on the 1ST
Least Daily Range 6° on the 28TH
Day of Month

Record of the Weather in New York for December, 1930
(Hourly Observation of the Relative Humidity Plotted on this Chart)


Greatest Daily Range 27° on the 1ST
Warmest Day 51° on the 9TH
Least Daily Range 2° on the 13TH
Coldest Day 7° on the 2ND
Day of Month

Record of the Weather in Chicago for December, 1930

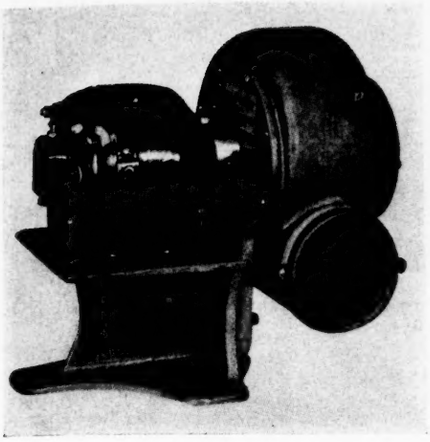

Warmest Day 65° on the 9TH
Coldest Day 18° on the 30TH
Least Daily Range 4° on the 14TH
Greatest Daily Range 24° on the 9TH
Day of Month

Record of the Weather in St. Louis for December, 1930

	New York	Boston	Pittsburgh	Chicago	St. Louis
Highest temperature, deg. F	56	58	50	51	65
Date of highest temperature	1	1	10	9	9
Lowest temperature, deg. F	9	4	8	7	18
Date of lowest temperature	16	16	2	2	30
Greatest daily range, deg. F	30	31	26	27	24
Date of greatest daily range	1	1	3	1	9
Least daily range, deg. F	6	5	3	2	4
Date of least daily range	28	4	8	13	14
Mean temperature for month, deg. F	35.0	33.2	31.6	29.7	35.8
Normal mean temperature for month, deg. F	35.0	32.5	34.2	28.8	34.9
Total precipitation, this month, inches	2.65	2.99	1.72	0.27	0.90
Total snowfall, this month, inches	4.2	7.7	2.8	3.2	0.5
Normal precipitation, this month, inches	3.62	3.45	2.86	2.04	2.21
Total wind movement, this month, miles	10,530	5852	7243	7066	8028
Average hourly wind velocity, miles	14.2	7.9	9.7	9.5	10.8
Prevailing direction of wind	N.W.	N.W.	W.	N.W.	N.W.
Number of clear days	6	12	3	6	13
Number of partly cloudy days	7	4	5	8	5
Number of cloudy days	18	15	23	17	13
Number of days with precipitation	12	8	13	6	3
Number of days with snowfall	3	2	6	5	1
Snow on ground, at end of month	0.1	1.0	0.5	None	None

Plotted from records especially compiled for HEATING AND VENTILATING by the United States Weather Bureau.
Heavy lines indicate temperatures in degrees F. Light lines indicate wind in miles per hour.
Broken lines indicate humidity in percentage from readings at 8 A.M., 12 M., and 8 P.M. Arrows fly with prevailing directions of wind.
S—clear, PC—partly cloudy, C—cloudy, R—rain, Sn—snow.

New Apparatus and Appliances



Nonoize Blower

Nonoize Blowers

A line of blowers designed to reduce the noise and vibration, has been announced under the name of Nonoize by the Therm-O-Vent Co., Inc., Philadelphia, Pa. A combination of special noise deadening materials is introduced between certain parts of the blower assembly so that any vibration generated is completely isolated and prevented from being transmitted to surrounding metallic objects. Air noises can be eliminated by the choice of a blower operating at low speeds. The blowers are available in 6 sizes, with capacities ranging from 55 to 2060 cu. ft. of air per min., free delivery.

Air Conditioning Newspaper Reel and Press Rooms

The Oklahoma Publishing Company has provided in its printing plant a humidification scheme which will effect a commodity saving per year of several times the installation cost, which amounted to approximately $5,000. The humidification represents something of a new departure, having at present only a single prototype, that in the original installation in the plant of the *Chicago Evening Post*. Frank D. Chase, Inc., Chicago, were the engineers and architects for the Oklahoma Publishing Company's plant.

The atomizing head employed is the direct result of research work and practical operation tests over a long period of years by the inventor, C. G. Flint, mechanical superintendent of the *Chicago Post*.

According to C. S. Millard, chief mechanical engineer for the architect, the essential feature of the Oklahoma installation is the use of the special atomizing heads in reel and press rooms. Each head is equipped with several nozzles through which steam and water are emitted in cloud-like vapor. Forty-two heads are used spaced from 10 to 20 ft. apart.

The heads are located in the ceiling in the reel room. Several units are placed in the areas devoted to the temporary storage of paper about to go to press. Other heads have aisle locations between the presses. In addition, one head is placed directly under the press in close proximity to the paper.

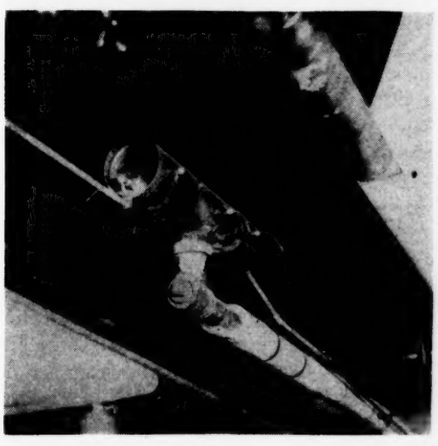
Aside from waste prevention by conserving paper from breakage and loss due to such breakage on modern high speed presses, the humidification process effects a distinct improvement in the appearance of the printed page. The type face appears much blacker and much more clear cut.

The amount of vapor discharged from the atomizing heads is subject to automatic regulation by means of a central humidity control equipment, according to Mr. Millard. Any desired humidity relationship can be kept constant. Recording instruments are installed in the reel room and furnish continuous tracings of conditions of temperature and humidity that obtain during printing processes.

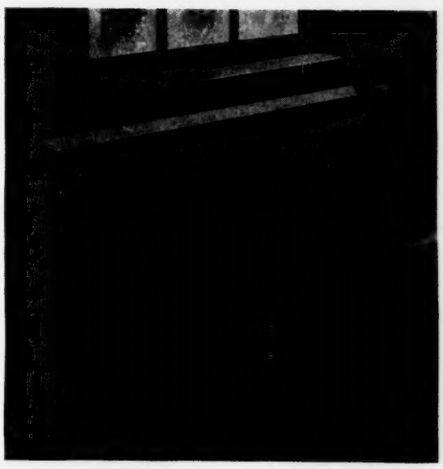
Each vapor head carries steam, water and drain connection. All steel piping is used. About 2,000 lin. ft. of piping are employed on the system with pipe sizes ranging from ¼ in. to 4 in. in diameter. It is an advantage in an installation of this kind to install piping after presses have been set.

When the heating system is in operation, a branch main supplies the necessary steam to the vapor heads through a pressure regulating valve. In the summer time, when the heating boilers are not in operation, steam supply is by an automatic boiler operating on natural gas. The water supply to heads that are located beneath the presses is automatically shut off when the presses are not running.

Space in the reel room provides for temporary storage of one or two days' supply of paper and gives this opportunity for preliminary absorption of certain amounts of moisture before the web is put on the presses. In the press room auxiliary humidifying equipment is carried on the columns opposite the folders. These Bahnson humidifiers are directed upon the folders. Their use averts trouble from static in the event that additional moisture is required in very dry weather.



Head for Humidifying Newspaper Plant



One of the Line of Arco Enclosures

Arco Radiator Enclosures

A line of radiator enclosures has been announced by the American Radiator Co., 40 West 40th St., New York. The line is complete in that enclosures are available for all types of American radiators, including the Murray radiator, Fantom radiator and the new Arco radiator, the latter being shown in the accompanying illustration, together with its enclosure. The new enclosures are attractive in appearance, easily installed and built so as to permit the greatest possible heating output from the radiator. They are also readily accessible for cleaning.

New Apparatus and Appliances

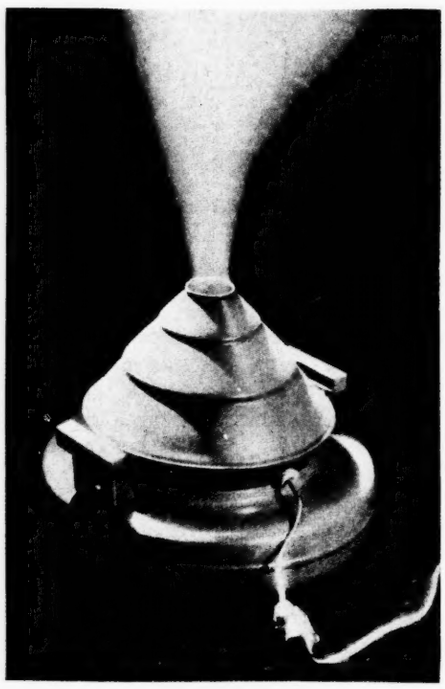
Ace Hygrometer and Thermostat

The Ace hygrometer and thermostat is being manufactured by Frederick Wackenhuth, 97 Lafayette St., Newark, N. J. It consists of a direct reading scale board; a wet and dry bulb thermometer; a U-shaped bi-metallic element for heat control, a U-shaped wet wick bi-metallic element for humidity control; adjustment screws and a water supply bottle.

Vital-Aire Humidifier

The Vital-Aire humidifier has been announced by The A. C. Gilbert Co., New Haven, Conn. Model B180 is a universal motor-driven device which operates on both alternating and direct current, 110 volts, and will handle from 5000 to 7500 cu. ft. of space. The induction motor-driven B181 humidifier operates on 110 volts, 60 cycle alternating current only, and will handle approximately 4000 to 5000 cu. ft.

For general purposes the manufacturer recommends the universal motor-driven B180 humidifier for offices and larger spaces. For the home, the induction motor-driven humidifier, B181, is to be used. There is no radio interference when using the B181. The device consumes between 40 and 50 watts. Both the B180 and B181 hold one gallon of water. If constantly operated, this is enough to last four hours in the B180 and six to eight hours in the B181. They retail at $49.50.



Vital-Aire Humidifier

Applying Holorib Roof Deck to the Paramount Theatre, Fort Wayne

Fenestra Holorib Insulated Roof Deck

Fenestra Holorib is the trade name of the steel roof deck manufactured by the Detroit Steel Products Co., Detroit, Mich. It forms a positive insulation for all kinds of buildings.

Holorib roof decks are fabricated of copper bearing steel sheets which range from 18 to 24 gauge, reinforced by self-contained triangular ribs rolled into the sheets longitudinally. These sheets are dip-coated and oven baked at the factory with a steel gray finish forming a touch protective film, which leaves no bare spots exposed to the weather. Sides and ends of the sheets dovetail and interlock securely to eliminate any possibility of leakage of asphalt.

Holorib is shipped from the factory in 18 in. widths and lengths as desired, in 3 in. variations up to 10 ft. 3 in. These sheets are securely anchored by welding the ribs to the purlins or by means of specially formed anchor clips. When used as a roof-deck, insulation such as cork, Celotex, JM Roofinsul, etc. is mopped in hot asphalt to the steel sheets and this material is covered with a built-up roof of rag or asbestos felt. The felts may then be covered with a final coating of gravel or slag if desired.

This type of roof is particularly advantageous for commercial and public buildings, such as factories, theatres, auditoriums, schools, hospitals, etc. It forms positive insulation, keeping the interior of the building cooler in the summer and reducing the fuel costs in the winter. It eliminates the troubles of condensation. A Holorib roof deck with insulation will weigh only from 4 to 7 lb. per sq. ft.

A Holorib roof can be installed in any weather. It is unaffected by atmospheric changes as there is practically no expansion or contraction, and it will stay water-tight.

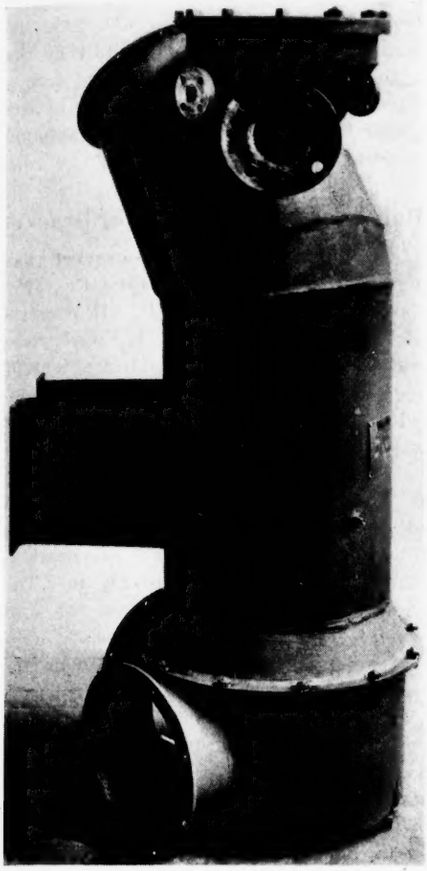
Method of Attaching Holorib to Channel Purlins by means of Standard Holorib Clips

Roper Announces Gas-Fired Furnace

A gas-fired warm-air furnace has been placed on the market by the Geo. D. Roper Corp., Rockford, Ill. The unit, which has been approved by the A.G.A. Testing Laboratory, is made in several sizes.

Each furnace is constructed with flues inside the jacket to furnish insulation. The heating unit is made of cast iron, and the furnaces are equipped with a humidifier. Accessories include a safety pilot, limit control and down-draft diverter. The jacket is finished in green.

New Apparatus and Appliances



Exhaust for Gasoline Tanker

Propeller Type Blower Exhausts Gases

The accompanying illustration shows a Westinghouse propeller-type blower of special design to be installed on a gasoline tanker to exhaust the gases from the pump room to the atmosphere. The blower is turbine driven and is rated at 6000 c.f.m. at 4 in. static pressure.

The entire unit is of welded steel construction and of simple design. The propeller is three-bladed and is driven by a single-wheel turbine having inside buckets milled in the wheel, through a steel shaft supported in grease-lubricated ball bearings.

Operation of this type of blower is free from objectionable noises, usually associated with vane-type blowers of this capacity and pressure.

Flettner Introduces Rotor Ventilator

A few years ago the Flettner rotorship aroused widespread interest in the Marine world. The designer, Anton Flettner, later turned his attention to ventilators, and as a result has succeeded in perfecting a device incorporating the rotor principle.

The Flettner ventilator consists of a metal ring fitted to the roof of a road vehicle, train, or other means of human locomotion, carrying a spindle running on ball bearings in an oil bath. The spindle passes through two superposed plates which revolve when air impinges upon them. A vacuum effect is created by a number of small valves under the lower plate, which draws out air from the interior of the vehicle. While the Flettner ventilator uses the air stream caused by the vehicle's motion, it continues to function when the vehicle is stationary and a wind is blowing, and its momentum is sufficient to carry it on for four minutes in stops in calm weather. Some thousands of these ventilators have been installed on the Berlin street cars, and it appeared on many of the cars shown at the last Berlin automobile show. It has now appeared on the English market, and there is a possibility that a stationary model may soon be introduced for house ventilation.

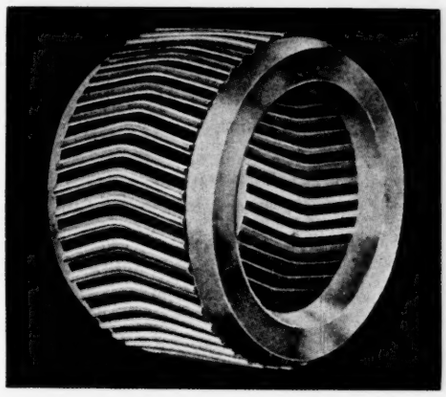
Ranarex CO_2 Indicator and Recorder

The Ranarex CO_2 indicator and recorder has been announced by the Permutit Co., 440 Fourth Ave., New York.

All working parts are readily accessible. The entire indicating and recording apparatus is attached to one plate, and the driving mechanism, including the motor, to another. Both plates may be removed by loosening a few wingscrews, their removal opening all gas passages over their entire length. Connection between the humidifier compartments and the measuring chambers is established by channels of ample cross-section.

Corrosion-resistant materials, such as bakelite and special alloy steels, are used for parts in contact with flue gas. The machine is inclosed in a dust-proof aluminum casing available for wall or panel mounting.

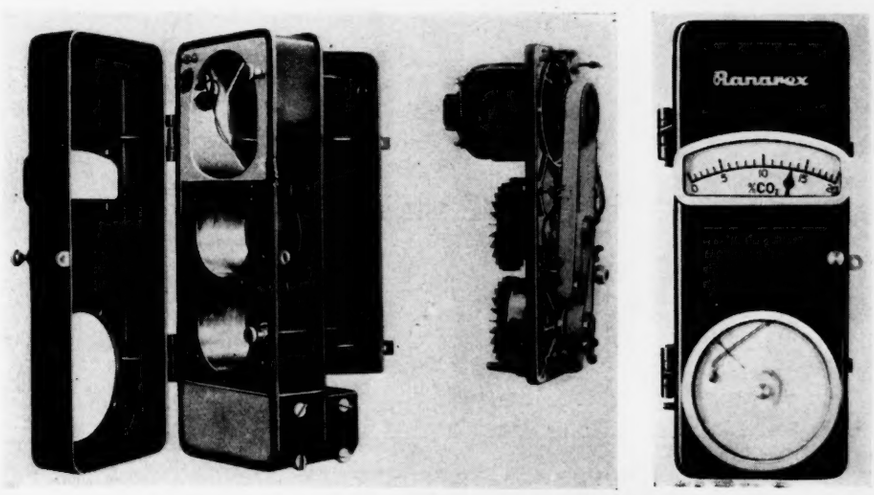
The standard 9-in. scale and circular chart 8 in. in diameter are graduated from 0% to 20% CO_2. Scales from 0% to 30% CO_2 for blast furnace gas and 0% to 40% for lime kilns are also available. The recording mechanism is driven interchangeably by a spring clock or an electric clock, as desired.



Herringbone Rotor for Sterling Motor

Sterling Motors with Herringbone Rotors

Announcement has been made by Sterling Electric Motors, Inc., 5401 Telegraph Road, Los Angeles, Calif., of a herringbone rotor for the company's line of motors. The new design provides noiseless operation, eliminates end thrust and the resulting wear on bearings. It is also said to do away with circulating currents, reduces exciting current, heating, eddy losses and iron losses.



Ranarex Wall Type CO_2 Recorder, Right; Left, Dust Proof Casing; Center, Indicating and Recording Mechanism

New Apparatus and Appliances

Thermotank Ventilation System

The Thermotank system of ventilation hitherto confined to the ventilation of passenger liners is now being associated with the ventilation of buildings and various industrial problems. The American agent is J. K. Ingham, 152 West 42nd St., New York.

The system embodies all the general principles of the conventional ventilating schemes but differs fundamentally in that the air is not delivered to the space ventilated through the ordinary register or ornamental louver at a low velocity, but through a special terminal outlet consisting of a spherical ball pierced by a tube, the ball being capable of adjustment to blow the air in any direction, as shown in the article on the ventilation of the motorship Asama Maru, in the January issue of HEATING AND VENTILATING.

Velocities of delivery from 30 to 50 ft. per sec. can be accomplished without any discomfort, the low velocities being used where perfect silence in operation is desirable. Where the air is delivered overhead, the motion is not physically perceptible, although a turbulent condition of the atmosphere is created which prevents stagnation, and, therefore, any tendency to temperature stratification.

The ability of the louver to be deflected downwards can, when warm weather necessitates this, create the effects of propeller and punkah fans.

An advantage of the Punkah louver delivery is that its high velocity is said to prevent any short circuiting between the delivery points and the exhaust grilles. In fact, satisfactory results are obtainable where the exhaust outlets are close to the Punkah louver itself.

The revolving Punkah louver consists of a Punkah louver ceiling fitting, which revolves automatically with the delivery of the air, the velocity being created by the reaction of small air nozzles without the use of any motor or complicated gear for drive. The high velocity gives a large radius of action and the rotation insures that every part is thoroughly ventilated.

In heating systems, especially in a ship or hotel, difficulty frequently arises in endeavoring to meet the varying requirements of different individuals. For meeting this difficulty, the Thermo-Reg Punkah louver has been designed. This consists of a terminal outlet incorporating the Punkah louver adjustable ball, and is suitable for fixing to twin trunks carrying hot and cold air, the rotation of the louver being so arranged as to control the relative quantities of hot and cold air being supplied.

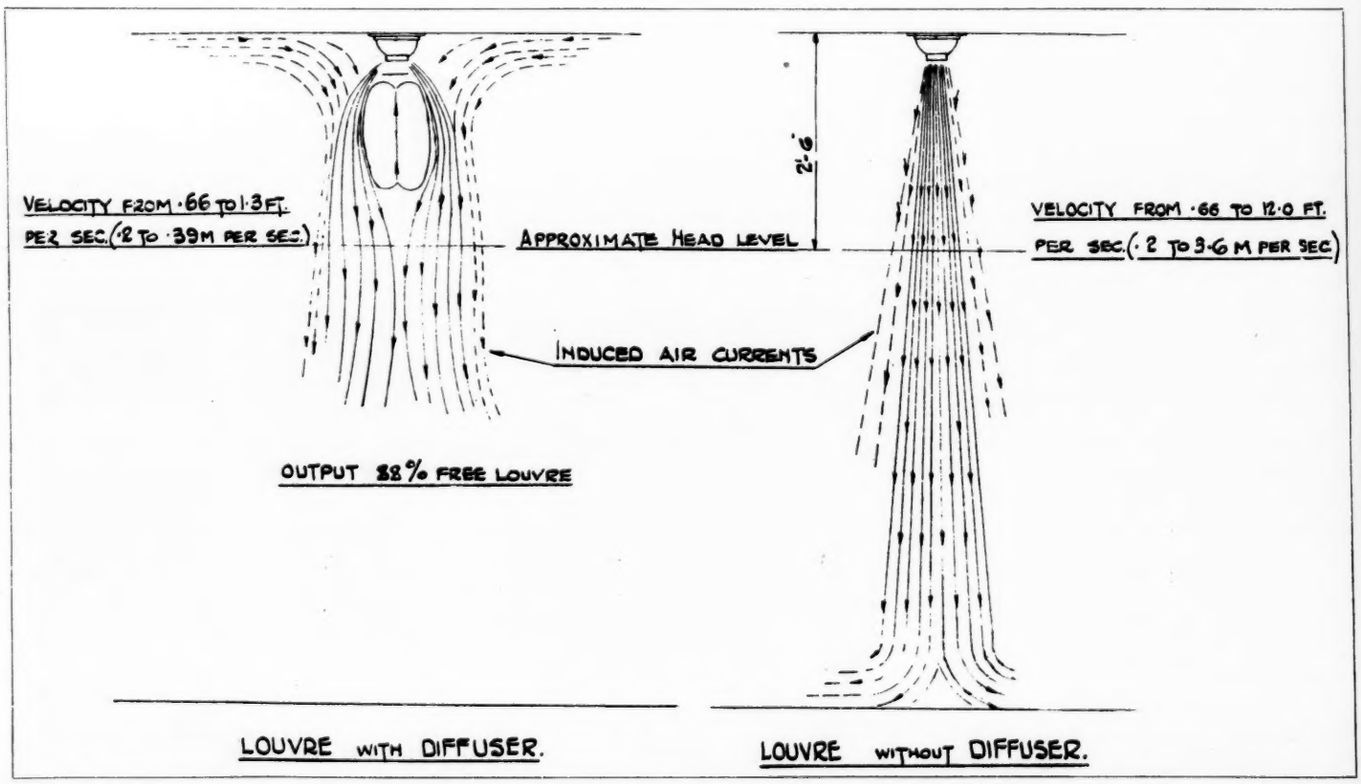
It occasionally arises that the louvers must be fitted on ceilings, in which position the turbulent discharge, while appreciated in hot weather conditions may be inconvenient during the heating season. To meet this condition, the patentees have designed a diffuser, which can be supplied with, or readily attached to their standard fittings, in the form of a flat plate which can be swung in front of the air jet. The disc is so designed that under this condition the high velocity stream is diffused into a stream of larger diameter and having approximately one tenth the maximum velocity of the free jet.

Paine Adjustable Radiator Bracket

An improved radiator bracket has been announced by the Paine Co., 2951 Carroll Ave., Chicago, Ill. It consists of three separate pieces: a wall piece which can be adjusted to the extent of 1½ in. vertically and 1¼ in. horizontally; the foot bracket which has an adjustability of 1 in. vertically; and the bolt and washers, the angular washer of which hooks into the slot at the top of the bracket as a means for holding the radiator firmly against the bracket. This slot is adjustable to 1¾ in. The radiator extends 1½ in. from the wall when set in its final position.

A separate foot bracket is made for each of the 3, 4, 5 and 6 tube radiators, while foot brackets for other than single column wall radiators are curved to conform to the shape of the bottom of the radiator.

The bracket is packed in individual carton, complete with lag screws and steel expansion shells for attachment to wood beams, and concrete or other solid materials.


VELOCITY FROM ·66 TO 1·3 FT. PER SEC. (·2 TO ·39M PER SEC.)
APPROXIMATE HEAD LEVEL
2'-0"
VELOCITY FROM ·66 TO 12·0 FT. PER SEC. (·2 TO 3·6 M PER SEC.)
INDUCED AIR CURRENTS
OUTPUT 88% FREE LOUVRE
LOUVRE WITH DIFFUSER.
LOUVRE WITHOUT DIFFUSER.

Comparison of Thermotank Punkah Louver With and Without Diffuser

"Although we have tried several other makes of automatic temperature control we have found none so accurate and dependable as Powers."

4 Reasons Why

Powers Temperature Regulation gives the biggest return on the money invested in automatic heat control:

[1] It often gives 15 to 20 years of Accurate and Dependable control without repairs of any kind.

[2] Greater Steam Economy due to the fact that thermostats do not get out of adjustment and permit overheating.

[3] Graduated Control, because of its greater accuracy, provides maximum comfort for occupants of rooms.

[4] A minimum of Service Calls and the trouble and annoyance connected with them.

THE POWERS REGULATOR CO.

40 years of specialization in automatic heat control

2720 Greenview Ave. **137 E. 46th Street**

CHICAGO — *Offices in 37 other cities* — NEW YORK

F. T. JOY, Vice President
W. E. MALLORY, Treasurer
C. A. MALLORY, President
H. B. MALLORY, Vice President
T. J. BOWEN, Secretary

RENOWNED SINCE 1823

THE MALLORY HAT CO.

FUR FELT & STRAW HATS

NEW YORK SALESROOM 234 FIFTH AVE. COR W. 27TH ST.

MAIN OFFICE AND FACTORY
DANBURY, CONN.

The Powers Regulator Company,
2720 Greenview Avenue,
Chicago, Illinois.

Gentlemen:-

Replying to your letter of February 1st regarding our experience with your automatic temperature control, since our original installation in 1919 we have added several units in different parts of our plant; our main building is completely equipped.

We consider the investment in this apparatus profitable for the following reasons: Fuel saving amounts to over 20 per cent; and Efficiency of workmen has been increased owing to air condition being much better for their health and comfort.

Your apparatus has also been a great help to us in maintaining proper temperature and humidity in our blowing and forming mill.

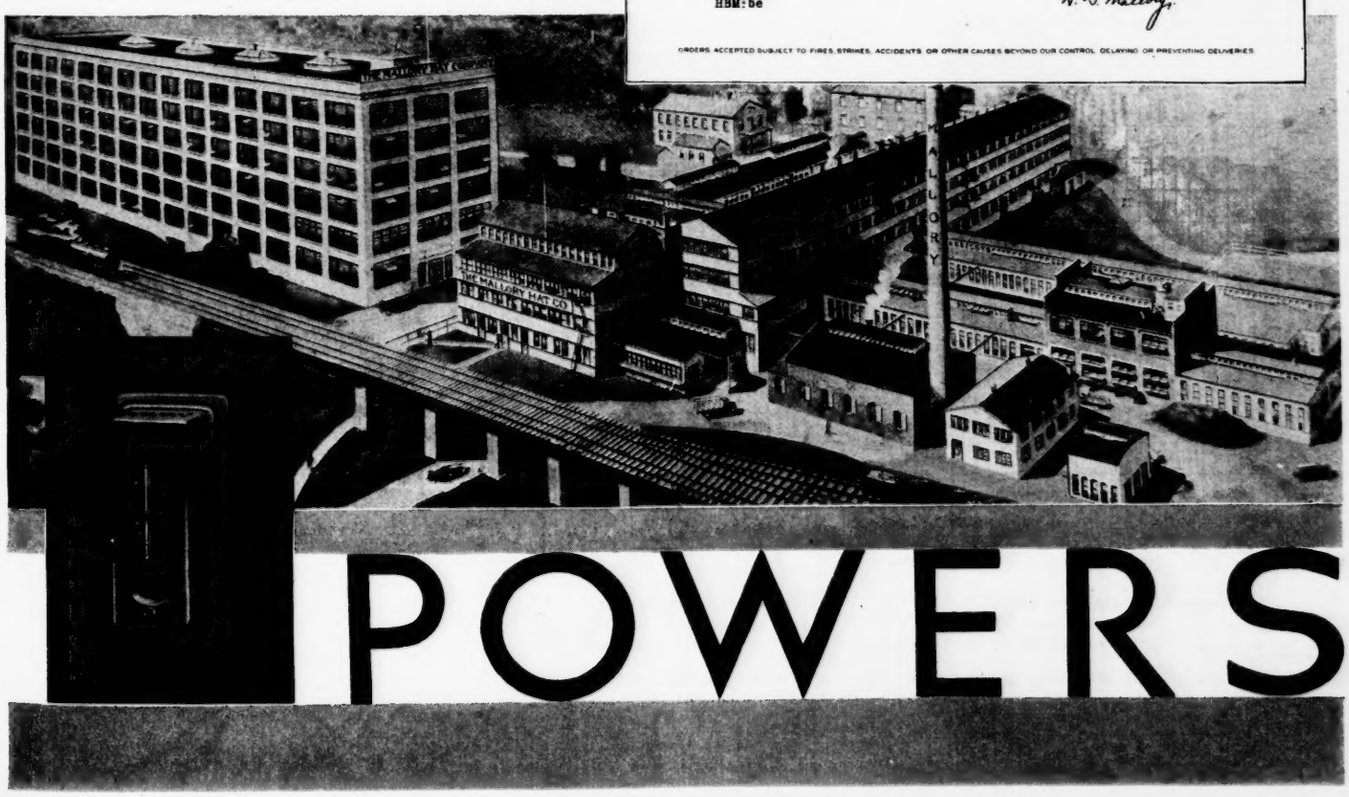
Although we have tried several other makes of automatic temperature control, we have found none so accurate and dependable as Powers

Very truly yours,
THE MALLORY HAT COMPANY

HBM:be

H. B. Mallory

ORDERS ACCEPTED SUBJECT TO FIRES, STRIKES, ACCIDENTS, OR OTHER CAUSES BEYOND OUR CONTROL, DELAYING OR PREVENTING DELIVERIES



POWERS

Degree Days and Unit Fuel Consumption in Typical Cities for December

UNIT fuel consumption figures given below must be corrected for efficiency, heat content of fuel other than that listed below, for radiation emitting other than 240 B.t.u. per hr., and for radiation calculated on a basis of other than zero to 70°.

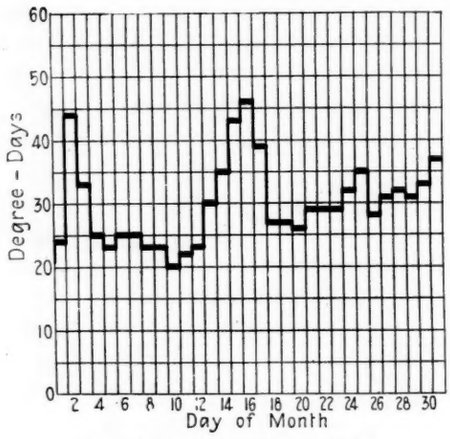
Coal is assumed as having a heat content of 12,000 B.t.u. per lb., gas, 1000 B.t.u. per cu. ft. and oil 140,000 B.t.u. per gal. To correct for other heating values, multiply the fuel consumption listed under the charts by the standard heat content just listed, and divide by the heat content in question.

To correct for efficiency, divide the fuel consumption by the efficiency of the heating installation.

For water radiation, multiply the fuel consumption, which is given for steam radiation, by 5/8.

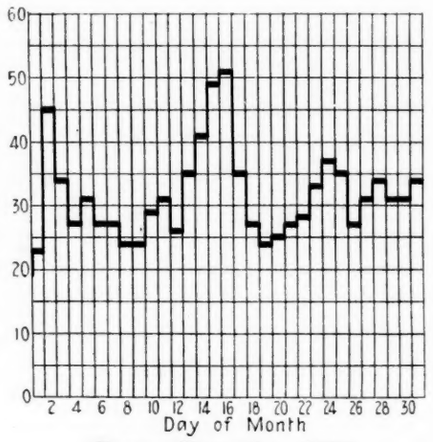
For radiation installed for any other outside temperatures than zero, multiply the fuel consumption given under the charts by 70°, divided by the quantity (70 minus the outside temperature used).

	Los Angeles	Baltimore	Philadelphia	New Orleans	Atlanta	Cleveland	Cincinnati	Detroit
Degree Days	154	871	876	392	790	1024	1002	1094
Degree Days to Date	270	1656	1675	584	1445	2089	2006	2316
Lbs. of Coal per Sq. Ft. of Radiation for December, 1930	1.232	6.968	7.008	3.136	6.320	8.192	8.016	8.752
Gals. of Oil per Sq. Ft. of Radiation for December, 1930	0.106	0.597	0.601	0.269	0.542	0.702	0.687	0.750
Cu. Ft. of Gas per Sq. Ft. of Radiation for December, 1930	14.78	83.62 (See Footnote)	84.10	37.63	75.84	98.30	96.19	105.02



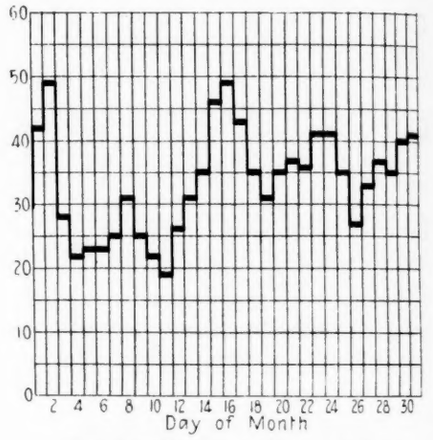
New York, December, 1930

Degree Days 929—to Date, 1834
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 7.432
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.637
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 89.18
(See Footnote)



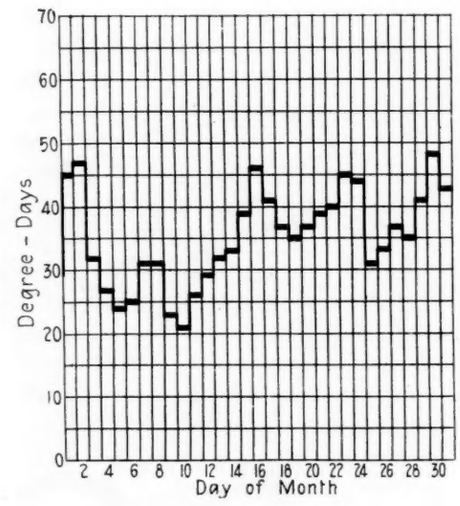
Boston, December, 1930

Degree Days 983—to Date, 1973
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 7.864
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.674
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 94.37
(See Footnote)



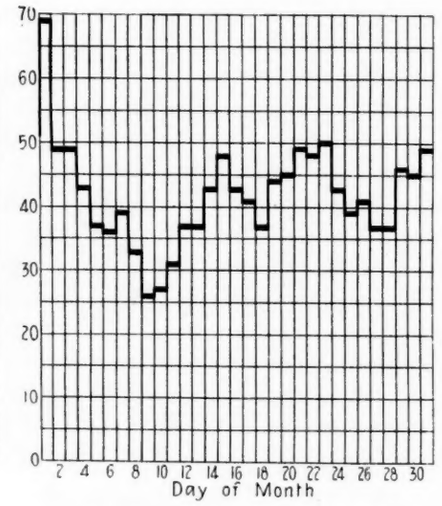
Pittsburgh, December, 1930

Degree Days 1043—to Date, 2134
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 8.344
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.715
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 100.13
(See Footnote)



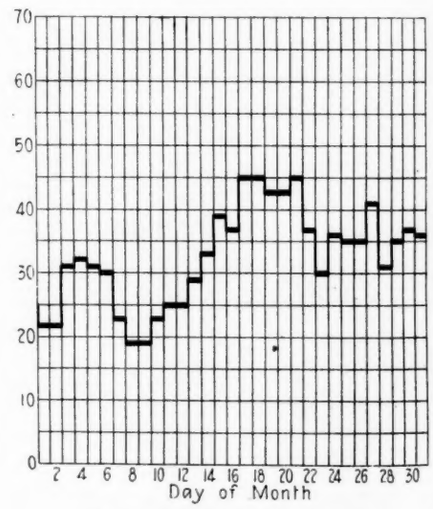
Chicago, December, 1930

Degree Days 1097—to Date, 2281
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 8.776
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.752
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 105.31
(See Footnote)



Minneapolis, December, 1930

Degree Days 1298—to Date, 2840
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 10.384
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.890
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 124.61
(See Footnote)



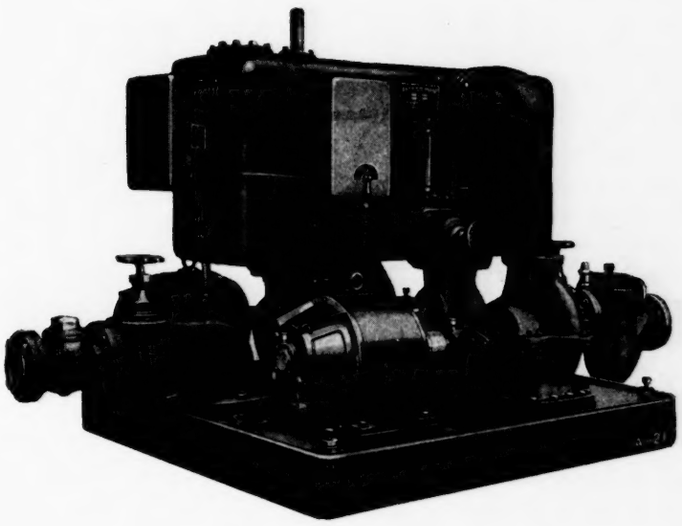
Denver, December, 1930

Degree Days 1014—to Date, 2298
Lbs. of Coal per Sq. Ft. of Radiation for this Month, 8.112
Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.695
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 97.34
(See Footnote)

These figures must be corrected for local conditions as explained above.

(*Continued on page* 108)

Facts



DURING 1931, this page will carry facts regarding vacuum pumps which should interest Owners, Architects, Engineers, Builders and Contractors.

These facts will explain how the vacuum system characteristics, which have a vital direct relationship to the pump duty, were all taken into consideration by the designers of the "AMES".

The "AMES" is not only distinctively different, but it has a built-in performance that means something well worth while to the user.

Write for Bulletins No. 552 and No. 553.

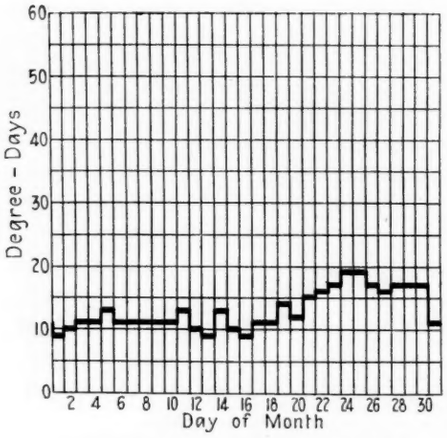
This advertisement is the first of the series.

Eleven more will follow.

AMES PUMP COMPANY, Inc.
30 CHURCH STREET NEW YORK CITY
Affiliated with
AMERICAN LOCOMOTIVE COMPANY

Degree Days for December, 1930

(*Continued from Page* 106)



San Francisco, December, 1930

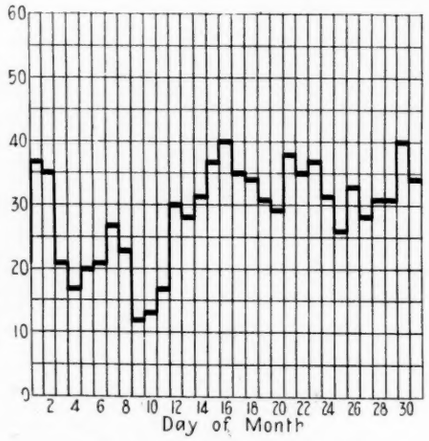
Degree Days 402—to Date, 797

Lbs. of Coal per Sq. Ft. of Radiation for this Month, 3.216

Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.276

Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 38.59

(See Footnote page 106)



St. Louis, December, 1930

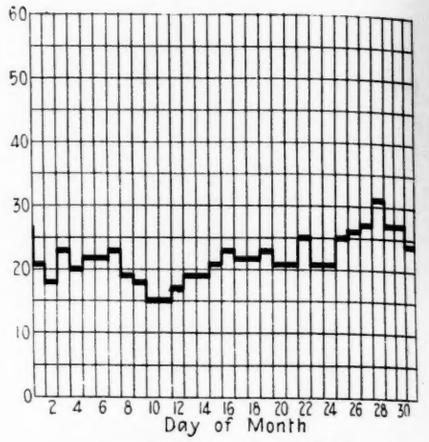
Degree Days 902—to Date, 1736

Lbs. of Coal per Sq. Ft. of Radiation for this Month, 7.216

Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.617

Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 86.59

(See Footnote page 106)



Seattle, December, 1930

Degree Days 678—to Date, 1889

Lbs. of Coal per Sq. Ft. of Radiation for this Month, 5.424

Gals. of Oil per Sq. Ft. of Radiation for this Month, 0.465

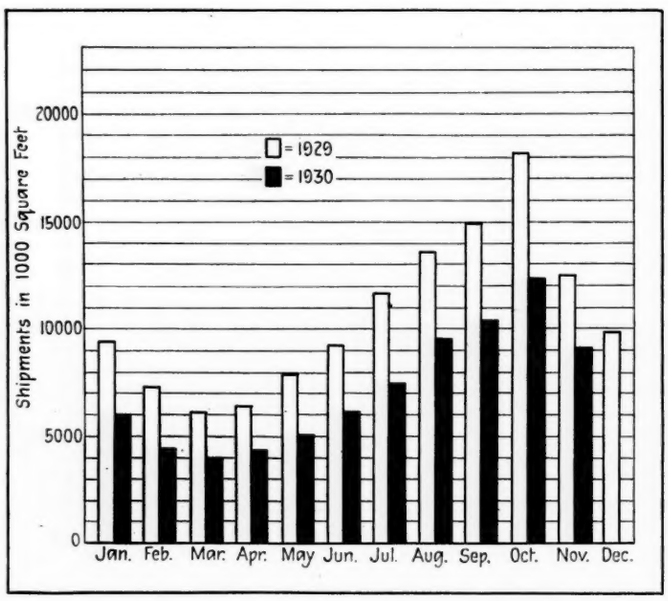
Cu. Ft. of Gas per Sq. Ft. of Radiation for this Month, 65.09

(See Footnote page 106)

HEATING AND VENTILATING

Statistical Service

Based on Data Supplied by the Department of Commerce

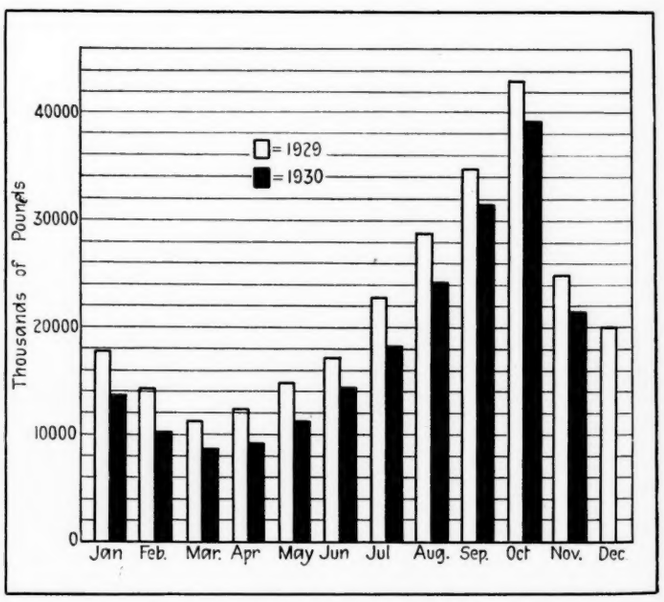


SHIPMENTS OF CAST IRON RADIATORS

(In 1000 Square Feet)

As Reported by 15 Manufacturers Representing over 90% of the Industry

November, 1930, 9,118; November, 1929, 12,501; decrease over last year, 27.1%; total for eleven months, 1930, 78,674; total for eleven months, 1929, 117,497; decrease over last year, 33%.



SHIPMENTS OF CAST IRON SQUARE BOILERS

(In Thousands of Pounds)

As Reported by 25 Manufacturers Representing 90% of the Industry

November, 1930, 21,480; November, 1929, 24,382; decrease over last year, 11.9%; total for eleven months, 1930, 202,627; total for eleven months, 1929, 242,313; decrease over last year, 16.4%.

(*Continued on Page* 110)

EMERSON
FAN DUTY
MOTORS

A complete line *from* one source *of* supply

YOUR production problems may vary—demanding a variety of sizes and types of motors for different applications.

But you need not vary the high standard of your motor equipment if you make Emerson your source of supply. The Emerson line is complete and interchangeable throughout to meet this variation in production requirements.

And don't overlook the all-important fact that Emerson's unusually wide experience in supplying fan duty motors for heating and ventilating devices may save you time and expense. Many of the fan motor problems regarded as particularly difficult have been perfectly solved in this line of fan duty motors.

Interchangeable

All Frames are Interchangeable for either Ball or Sleeve Bearings

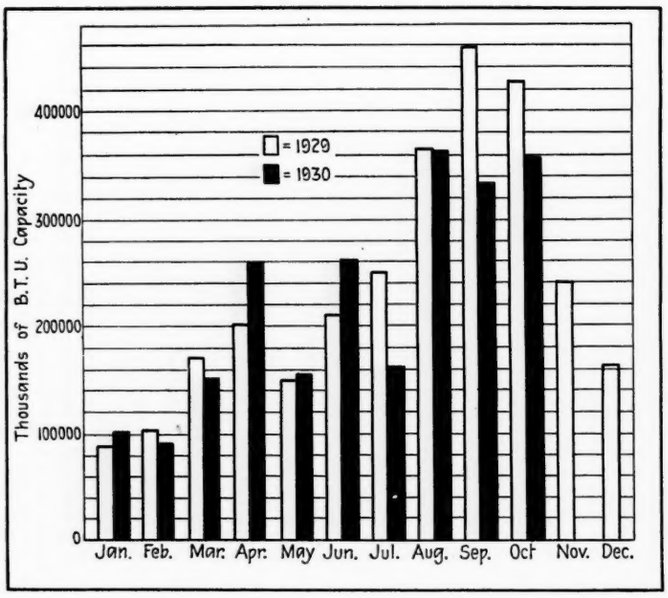
Single Phase, Single-Speed
Single Phase, 2-Speed
(Speed ratio 2 to 1)
(Speed ratio 3 to 1)
Single Phase, 3-Speed Capacitor
Polyphase, Single-Speed
Polyphase, 3-Speed Bank Control
DC Single or Variable Speed
1/30 to 1/2 h.p.

THE EMERSON ELECTRIC MFG. CO.
2018 Washington Ave., Saint Louis
806 W. Washington Blvd., Chicago
155 Sixth Ave., New York City

Write for Bulletin

Statistical Service

(*Continued from Page* 108)

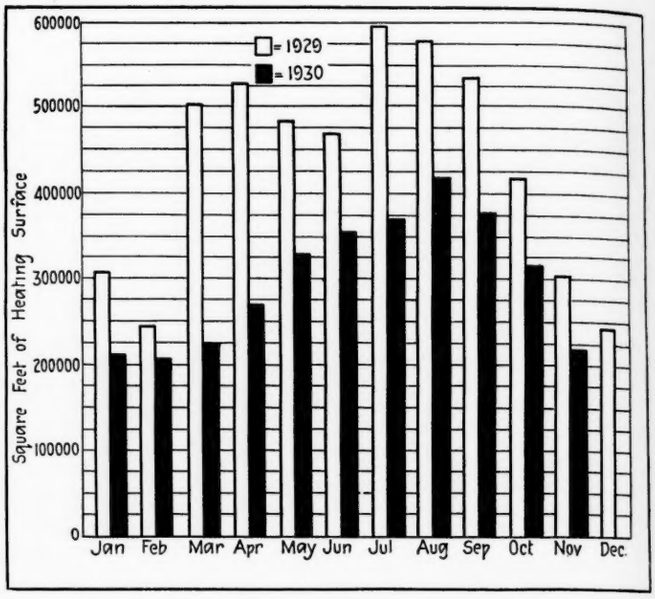


SHIPMENTS OF GAS-FIRED BOILERS

(In Thousands of B.t.u. Capacity)

As Reported by 8 Manufacturers Representing 75% of the Industry

October, 1930, 359,206; October, 1929, 427,629; decrease over last year, 16%; total for ten months, 1930, 2,236,581; total for ten months, 1929, 2,432,981; decrease over last year, 8.08%.

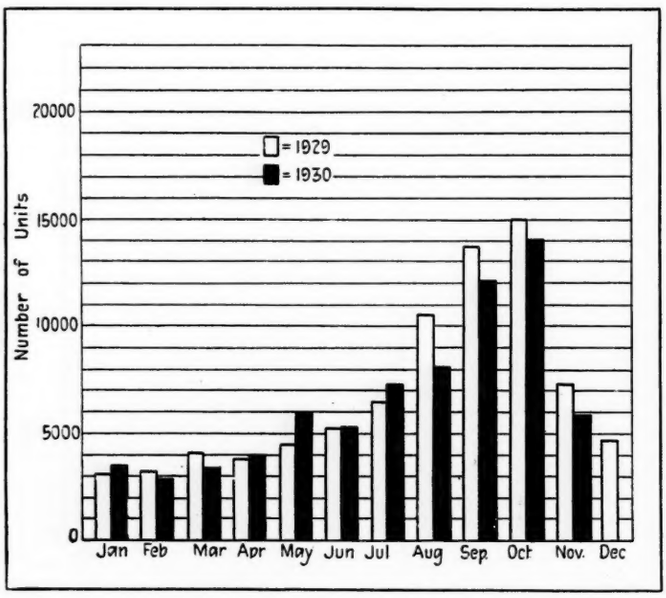


NEW ORDERS FOR STEEL HEATING BOILERS

(In Square Feet of Heating Surfaces)

As Reported by Firms Making Up the Greater Part of the Industry

November, 1930, 218,865; November, 1929, 306,867; decrease over last year, 28.6%; total for eleven months, 1930, 3,308,969; total for eleven months, 1929, 4,987,780; decrease over last year, 33.6%.

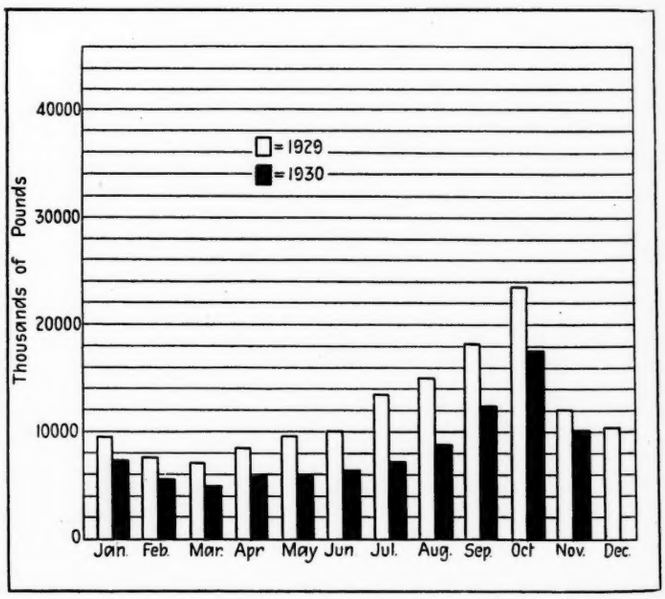


SHIPMENTS OF OIL BURNERS

(In Number of Units)

As Reported by 50 Manufacturers Representing 60% of the Industry

November, 1930, 5,880; November, 1929, 7,213; decrease over last year, 18.5%; total for eleven months, 1930, 72,095; total for eleven months, 1929, 76,966; decrease over last year, 6.34%.



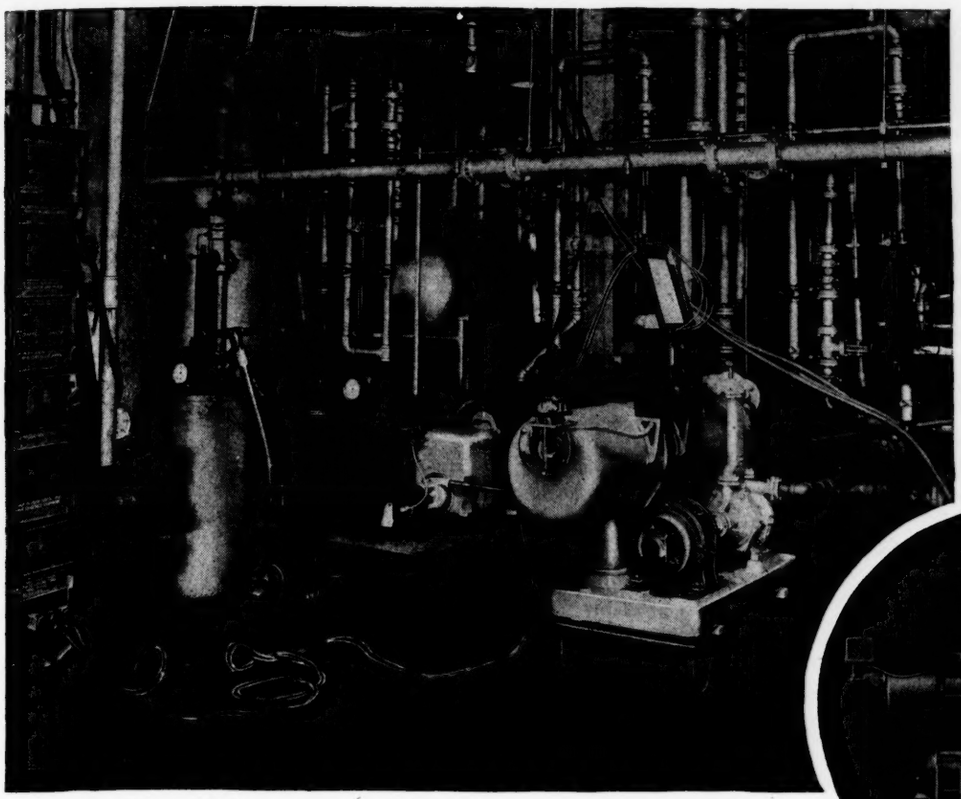
SHIPMENTS OF CAST IRON ROUND BOILERS

(In Thousands of Pounds)

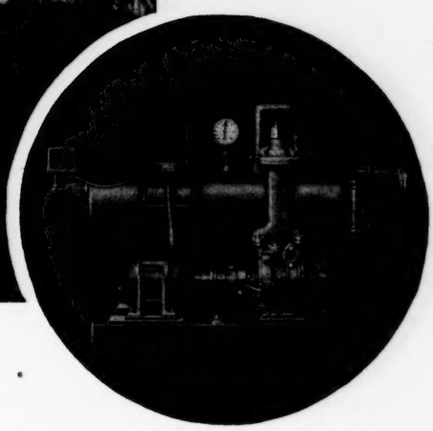
As Reported by 25 Manufacturers Representing 90% of the Industry

November, 1930, 10,041; November, 1929, 12,162; decrease over last year, 17.5%; total for eleven months, 1930, 91,827; total for eleven months, 1929, 135,502; decrease over last year, 32.2%.

Here pump capacity



Jennings Vacuum Heating Pumps are furnished in capacities of 4 to 400 g. p. m. of water and 3 to 171 cu. ft. per min. of air. For serving up to 300,000 sq. ft. equivalent direct radiation. Write for Bulletin 85



General view of laboratory in Nash factory where Jennings Pumps are tested.

is proven by test

BEFORE specifying a vacuum heating pump, there is one all-important fact to make sure of. This is whether the pump selected has ample capacity to handle the job.

If the pump you specify is a Jennings, you can be positive that it has.

The air and water capacities of every Jennings Heating Pump are determined for you by actual test at the Nash factory. These tests are conducted under the same vacuum and the same pressure conditions as are to prevail in the actual installation. These tests are your guarantee of satisfactory heating pump performance.

Due to the unique Jennings principle of operation, air and water are handled separately. Air capacity is not affected by the volume of water being pumped. Water capacity remains constant when air is being handled. Combined air and water capacity is equal to maximum air capacity plus maximum water capacity.

Jennings Pumps

THE NASH ENGINEERING CO 81 WILSON ROAD, SOUTH NORWALK, CONN.